JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XVII. NOVEMBER, 1905. No. 4.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY

(Under whose supervision this number is published).

Rear Admiral GEORGE W. BAIRD, U. S. N., Retired. Commander J. K. BARTON, U. S. N.

Captain A. F. DIXON, U. S. N. Commander W. M. PARKS, U. S. N.

Lieutenant CHAS. K. MALLORY, U. S. N., Retired.

THE RATEAU STEAM-TURBINE AND ITS APPLICATIONS.

BY M. J. REY.

Translated from the April, 1904, issue of the Memoirs and Compte Rendu of the Works of the Society of Civil Engineers of France, by Chief Engineer B. F. ISHERWOOD, United States Navy, Honorary Member.

The substitution of the steam-turbine for the reciprocating piston steam-engine, which has been realized in the present epoch, is the most important fact in the history of steam-motors, if not the greatest step in the progress of the mechanical arts, since the memorable invention of Watts.

Numerous precursors, sagacious inventors, penetrating intellects, of the early years of the XIX century, foresaw the future reserved for the rotative movement and continuous flow as opposed to the intermittent flow and alternate movement alone employed up to almost the present time.

In proof, there can be cited the name of Tournaire, Inspector General of Mines in France, whose theoretical description of a steam-turbine with multiple reaction wheels, addressed in 1853 to l'Académie des Sciences, is startling when compared with the engine constructed by Parsons in England 34 years

afterwards. Also the name of Pilbrow, who, in a patent taken out by him in 1843, described a steam-wheel furnished with convergent-divergent distributors on the principle embodied later in the well known steam-turbine of Laval in Sweden.

To realize and to utilize the steam-turbine, certain conditions are indispensable:

1st. The exact knowledge of the physical properties of steam, as well as of the thermo-dynamic laws enabling the calculation to be made of the velocity of the flow of steam, and of the total work it can do for a given fall of pressure.

2d. The possibility of using metals of great strength and perfect homogeneity, capable of sustaining the enormous strains caused by centrifugal force in revolving bodies at the high angular velocities produced.

3d. A new mode of transmitting the power of the turbine to the varied receiving machines used in industrial operations that will utilize, without notable loss, its great velocity.

The progress of the physical sciences, especially the progress of metallurgy, and the discovery of the electrical generatrices and motors of induction, have now given a practical solution of this great problem, and have rendered possible this revolution in industrial mechanics.

At the passing moment, to say that the steam-turbine will be the steam-engine of the future, would not be correct: it is already the steam-engine of the present. Several hundreds of thousands of horsepowers of it are now either in use or in construction; and there is no temerity in the statement that with the exception of locomotives, lifting engines, rolling engines, and, in a certain sense, marine engines, there will be no more building of reciprocating steam-engines.

There is no intention to here give a resumé, however succinct, of the efforts which have been made by engineers and specialists to produce the steam-turbine of these latter years. To treat the subject as a whole, the numerous patents of the precursors* would have to be passed in review and examined;

*See, notably, the interesting investigation: *Roues et Turbines à Vapeur*, published by Mr. Sosnowsky in le Bulletin de la Society d'Encouragement à' l'Industrie Nationale. 5th Series, tome II, page 1153 and following.

certainly, several hundred of them at the present time taken out in the different principal countries during the last fifteen years.

The intention is much more modest; it will be restricted to a monograph of the steam-turbine created by Mr. Rateau, Engineer in the Corps of Mines, in collaboration with the firm of Sautter, Harlé & Co., which has made from the beginning all the investigations and all the experiments concerning the new type of engine.

In order to render the question precise, a rational classification is indispensable of the different systems of steam-turbines, so as to be able to state the exact category to which the Rateau turbine belongs, its principle of construction, its mode of functioning and its technical advantages.

Previous to making this classification, a relation of some historical facts regarding the invention itself, would seem to be useful.

HISTORICAL FACTS RELATING TO THE RATEAU TURBINE.

The principal works of Mr. Rateau on steam-turbines, go back to the year 1894, in which he commenced the construction of a steam-turbine with one rotating wheel for driving dynamos.

But, before this epoch, Mr. Rateau had been occupied with the question of steam-turbines, and there will be found in the monthly comptes rendus of the Societé de l'Industrie Minerale (seances of March 1st and 12th April, 1890), an interesting communication from that engineer concerning the Parsons turbine, as well as a discussion of the use of steam in mass motion or under pressure.

The fact is important that Mr. Rateau described at that epoch, the construction of steam-turbines with wheels and distributors moving in opposite directions, an idea which has recently been taken up again by different inventors.

The investigations of the steam-turbine of one wheel were pursued during the years 1895 and 1896, at Saint-Etienne and at Paris. After having tried successively the centrifugal method and the centripetal method, they resulted in the adop-

tion of groups of electrogens of one or two dynamos with helicoidal wheels of the Pelton system, of which a small number were constructed and are still in use.

Mr. Rateau very soon perceived the disadvantages of the one wheel turbine, and the advantages of a system of multiple wheels, the working steam acting by impulsion instead of by the method of reaction employed by Parsons. His ideas took definite form towards the year 1897, and during that epoch he conceived the principle of his helicoidal multicellular turbine of impulsion, the merits of which will be shown later on. The following year the firm of Sautter-Harlé commenced the construction of a steam turbine of 900 horsepowers based on this principle.

Of the numerous difficulties encountered in the course of the construction, and developed by the first experiments, only a few need be recalled: they were those of obtaining the perfect equilibrium required in the rotating pieces; those of effecting the automatic lubrication of the pillow-blocks; those of the packing of the stuffing-boxes of the shaft; those of the regulation of the velocity; those of the best arrangement for reducing the friction of the discs upon the steam; and those of other details that would require too much space for description; demanding a great deal of thought and time to overcome them and perfect the practical working of the engine.

A first unit of 1,000 horsepowers was intended for the Exposition of 1900. It was not ready in time, and the first experiments were made with it after the closing of the Exposition.

While the turbine was being created, the generator of electricity that it was to operate had also to be created. The triphase current was attacked, and there was constructed a first alternator of iron revolving with a power of 700 kilowatts.

From the year 1901 the progress was rapid.

The employment of low pressure turbines also increased the field of action of these new machines. At the present time the firm of Sautter-Harlé have either delivered or have in construction more than 17,000 horsepowers of turbines, and this figure is rapidly increasing.

Before terminating what was to be here said about the theoretical work of Mr. Rateau, there must be added that he has deposited in the Academy, a dossier containing all his researches up to that period on steam-turbines.*

CLASSIFICATION OF DIFFERENT SYSTEMS OF STEAM-TURBINES.

Steam-turbines constitute a division of the general class of engines in which a fluid does external work by acting upon one or more rotating wheels by means of buckets, chambers, or passages made in or upon them. [Bucketing.]

In the steam-turbine the rotating wheel is pushed by the motor-fluid, the appropriate direction of which is given by distributing passages or canals.

The steam-turbine, or an element of it, is consequently composed of two fundamental organs: the rotating wheel and the fixed distributor.

The rotating wheel can be formed in a number of ways. According to the case, it may be one massive wheel in a single piece dressed from a block of metal and having its buckets cut in it. Or it may be of plate iron with its buckets affixed to it. Or it may be a solid wheel on the periphery of which its buckets made in two separate bands may be shrunk on.

The inventors have free license, and many methods are possible; but, in reality, there are only two prevailing types, namely, the light wheel of plate iron with buckets applied,

---

*The different notes which this dossier contained, were those already published, and others not yet edited. They relate to all the interesting subjects concerned in steam-turbines. Their names are as follows:

1st. *Report on Steam-Turbines* made to the International Congress of Applied Mechanics, in 1900.

2d. *Treatise on Turbo-machines* (published in 1900).

3d. *Abacus of the theoretical consumptions of a steam-engine* (published in the Annales des Mines of 1897).

4th. *Resistance of Revolving Bodies* (published in the Bulletin de la Societé de l'Industrie Minerale of 1890).

5th. *Apparatus for measuring the humidity of steam* (Annales des Mines, 1898).

6th. *Memoir of Experimental Researches concerning the Efflux of Steam* (inedited, since published in the Annales des Mines, 1902).

7th. *Theory of Steam-Turbines* (inedited, since published in the Revue de Mechanique, 1903).

8th. *Experiments on the impulsion of steam* (inedited).

9th. *Experiments on Steam-Turbines* (inedited).

10th. *Experiments on a high pressure centrifugal ventilator* (inedited, since published by the Societé d'Encouragement, and in the Bulletin de l'Industrie Minérale, November, 1901).

and the solid wheel with its buckets cut out of the same block.

A third method of construction can be adopted, consisting in the suppression of the body, properly so called, of each wheel, and mounting the different bands of buckets upon a single drum.

The distributor is formed, as in the case of hydraulic turbines, by curved directrices; or it is composed simply of one or of several tuyeres of circular cross section, calculated according to the properties of the expansion of elastic fluids.

The construction of distributors can be very much varied; sometimes canals or passages are cut into the mass of the metal; or are formed, on the contrary, by separate small pieces of metal applied to it; and, according as the distribution is made over all the periphery, or over a fraction of it, the construction can be varied in many ways.

The classification of steam-turbines can be made according to three kinds of consideration:

1st. According to the number of rotating wheels composing the turbine.

2d. According to the route followed by the motor-fluid.

3d. According to the mode of functioning of the motor-fluid.

These three categories will now be successively passed in review, and examined in order to ascertain the different kinds of turbine represented by each in its turn.

NUMBER OF ROTATING WHEELS.

The classification of the number of rotating wheels, comprises two kinds:

The turbine with a single rotating wheel;

The turbine with multiple rotating wheels.

The turbine with a single rotating wheel is divided, in its turn, into a turbine with the buckets applied, and a turbine with the buckets cut out of the same block of metal as the wheel.

Among the single wheel turbines with applied buckets may be particularly cited the Laval turbine and the Seger turbine.

The Rateau turbine of one wheel, constructed in 1894, has the buckets cut out of the mass of the metal, the whole wheel being in a single piece. It is the same with the Riedler-Stumpf turbine of which so much has been said in Germany during the last two years.

The multiple wheel turbine comprises the following three kinds:

The drum turbine;

The turbine with groups of wheels;

The multicellular turbine.

Among the drum turbines, the best known and the most extensively used is the Parsons turbine.

The turbines with groups of wheels—also termed turbines with speed in stages—appear for the first time with the Curtis turbine.

As for the multicellular turbines, the best known, actually in use, is the Rateau turbine.

ROUTE FOLLOWED BY THE MOTOR-FLUID.

The route followed by the motor-fluid in the rotating wheel can be determined in two principal ways.

In the first case, the motor-fluid traverses the wheel parallel to its axis. The axial or helicoidal turbines are thus obtained.

During this route the fluid continues sensibly at the same distance from the axis of the wheel.

In the second case, the motor-fluid moves in a plane perpendicular to the axis of the wheel. The radial turbines are thus obtained, and they comprise two kinds. The centrifugal, in which the fluid moves from the axis. The centripetal, in which the fluid moves towards the axis. Finally, these two modes—the axial and the radial—can be combined by causing the fluid to move obliquely in relation to the axis, and in this way will be obtained the helico-centrifugal or helico-centripetal turbine.

As examples of the helicoidal turbine, and of the axial tur-

bine, there may be cited all the steam-turbines employed in practice.

The mode of centrifugal route, or of centripetal route, has given, notwithstanding numerous trials, but very mediocre results.

The radial turbines are a great deal more complicated in construction, and they do not give any advantage either in fuel economy or in mechanical functionment over the helicoidal turbines.

MODE OF FUNCTIONING OF THE MOTOR-FLUID.

The third category for the classification of steam-turbines is based on the mode of the functioning of the motor-fluid.

As, in the case of hydraulic turbines, all steam turbines can be made to function either by reaction or by impulsion.

The first group, or reaction turbines, comprehends notably the Parsons turbine and the Westinghouse turbine, which is only a variant of it.

In the second group are found the turbine of Laval, the turbine of Rateau, the turbine of Curtis, the turbine of Riedler, and the turbines of many others.

I.—CLASSIFICATION BY THE NUMBER OF WHEELS.

There will now be passed in review, a certain number of examples chosen from the best known steam-turbines, in order to illustrate the classification made, and to enable it to be better comprehended.

TURBINE OF ONE WHEEL.

Figure 1 represents the well known helicoidal wheel of Laval's steam-turbine.

There will be remarked the distributors formed of converging-diverging tuyeres placed obliquely to the wheel, which is formed of one solid piece with applied buckets shaped like the tail of a swallow, and held by a metallic felly.

By using a special steel employed by the firm Bréguet in the construction of this wheel, its peripheral velocity can, in practice, be pushed up to 1,150 feet per second; in some experiments 1,300 feet were obtained.

There may here be recalled that the Laval wheel is mounted upon a shaft of small diameter whose flexibility allows the axis of inertia of the wheel to follow the geometrical axis of the wheel notwithstanding the small defects of equilibrium impossible to be suppressed in practice.

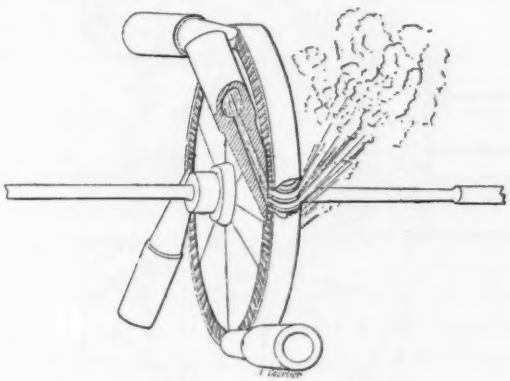


Fig. 1.—LAVAL TURBINE.

Figure 1, Plate I, shows the helicoidal wheel of a Rateau turbine of one wheel on the Pelton system.

This system, patented in France in 1894, is constituted essentially of a wheel with its buckets cut out of a single block, instead of the buckets being made separately and attached. This construction assures greater strength to the rotating wheel. The wear by the friction of the buckets does not cause any more inconvenience, because their thickness is much greater. There is no more danger of the wheel being thus thrown out of equilibrium by the suppression or the diminution of one or of several buckets.

With a wheel of this construction of 1 foot diameter, it is possible to make 24,000 revolutions per minute which represents a peripheral velocity of, say, 75,400 feet per minute.

As has been remarked, the buckets have the form of those on the Pelton wheel; consequently, the distribution of the steam is made diametrically, and is perfectly symmetrical as

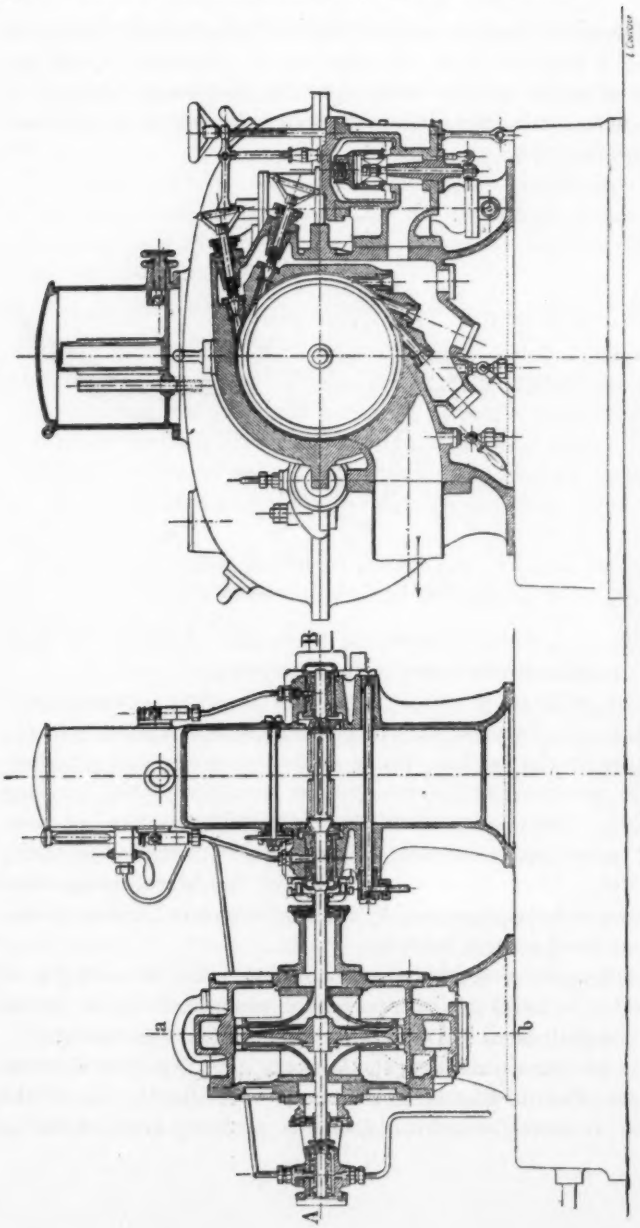


Fig. 2.—Turbine of One Wheel, Rateau System.

regards the wheel, which is thus free from any longitudinal or axial push.

The combination of the wheel and the distributors, which latter are formed of diametrically placed tuyeres, is shown in Figure 2, which represents an electrogen group of 40 kilowatts power at 15,000 revolutions per minute, constructed by the firm of Sautter-Harlé and Company in 1895.

Several examples of these turbines are in operation at the present time.

Figure 2, Plate I, represents one of them. The distribution by diametrically opposite tuyeres is plainly visible. The shaft of the turbine carries a pinion operating by two sets of helicoidal gearing, the two shafts of the dynamos.

This arrangement is the same as that which is employed in the Laval turbine. It has also the same disadvantages in the transmission of the power from the turbine to the dynamos.

During these last years the construction of the wheel with buckets according to the Pelton system and designed by Mr. Rateau, has been again put in practice in Germany, by Professors Riedler and Stumpf for a one wheel turbine constructed by the Allzemeine Elektricitats Gesellschaft.

Figure 3, Plate I, represents a Riedler and Stumpf wheel for a turbine of 2,000 horsepowers. This figure is extracted, as well as the following ones, from a Memoir by Riedler to the German Technical Society for the construction of vessels.

The distributors employed for the Riedler and Stumpf turbine, are represented in section and in external view in Figure 4, Plate I.

Attention is called to the great number of tuyeres required to form a distributor for the Riedler and Stumpf wheel. For great powers, there must be delivered on the same wheel a correspondingly large weight of steam, and it is impossible to apply to the periphery of the wheel larger tuyeres and fewer in number, because their length would become too great.

Inventors soon comprehended that a turbine with but one wheel, could have only a mediocre development of power, and they have adopted for it two wheels the general action of

which is represented for a turbine of 2,000 horsepowers in Figure 3.

There will be described later, the different causes that affect the mechanical rendering in a steam-turbine, but there can be easily shown immediately the advantage of multiplying the number of wheels to fraction the fall of the steam-pressure.

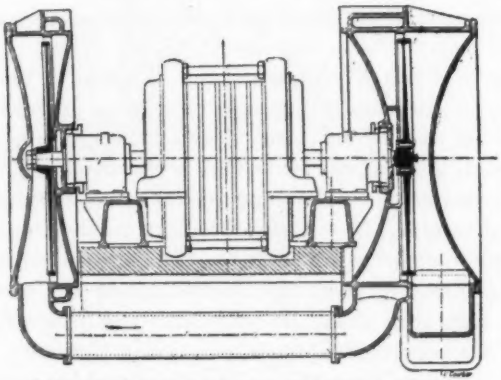


Fig. 3.—TURBINE OF TWO WHEELS, RIEDLER.

The mechanical rendering of a steam-turbine depends essentially on the ratio of the peripheral speed of its rotating wheel to the absolute speed of the steam at the moment of introduction into the turbine. With the considerable speeds of elastic fluids that result from a fall in their pressure, there would be required either enormous peripheral speed for the rotating wheel of the turbine or a fractionment of the total fall of pressure into a series of smaller falls.

If steam of the effective pressure of 142.25 pounds per square inch be used, expanding into a vacuum of 25.58 inches of mercury, the speed of the efflux for saturated steam will be 3,724 feet.

To obtain an acceptable mechanical rendering with a one wheel turbine, the peripheral speed of the wheel must exceed 1,312 feet, a limit very difficult to attain for wheels of large diameter. Thus, for a turbine of 1,000 kilowatts, for example,

making 1,500 revolutions per minute, the diameter would have to be at least 16.4 feet for the peripheral speed required. With such dimensions, the construction of a wheel with the exactness needed for the perfect equilibrium of all its parts, would be nearly impossible. Therefore, it is not astonishing that the inventors of single wheel turbines have successively increased the number of their elements.

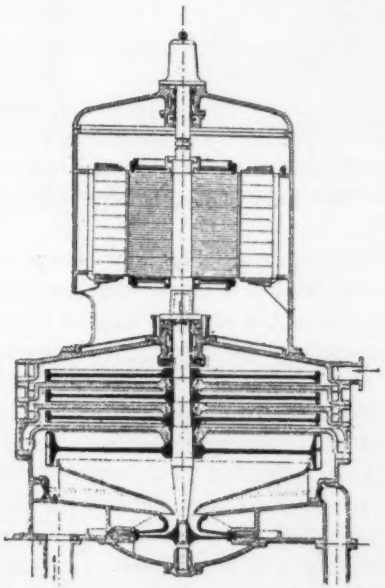


Fig. 4.—Riedler Turbine with Four Wheels.

Figure 4 represents a Riedler and Stumpf turbine of 500 kilowatts, in four successive chutes of steam.

With the exception of turbines of sufficiently small powers to allow the use of speed-reducing gearing, the application of turbines of one wheel is not considered practicable, except for pumping engines having centrifugal pumps, and for ventilating engines, both delivering at a great height, the utilization in these cases being immediate at extremely great rotary speeds.

Thus, the turbines with multiple-wheels are the only ones of a truly general application.

TURBINES WITH MULTIPLE-WHEELS.

DRUM-TURBINES WITH ROTATING CIRCLET OF BUCKETS.

The characteristic of this kind of turbine, is the employment of a rotating drum carrying a circlet of buckets similar to each other.

The ensemble of a turbine thus constituted consists of several drums of different diameters, each surmounted by a certain number of circlets of buckets.

The prototype of the drum-turbine is the turbine of Parsons.

The Parsons turbine of 1,500 horsepowers furnished to the City of Elberfield, and with which experiments giving astonishing results have been made, is represented in Figure 5, Plate II.

This turbine operates an alternator of 1,000 kilowatts, revolving with the speed of 1,500 revolutions per minute.

The construction of the rotating band of buckets and of the distributor, as practiced by Parsons, is represented in Figure 6, Plate II.

With the exception of dimensions, this kind of construction is generalized for this variety of turbine.

The distributor is formed of a band of curved channels fixed in an exterior envelope. This band is keyed to the drum. The height of these channels varies, naturally, with the weight and volume of steam used.

Figure 7, Plate II, shows an interior view of a Parsons turbine, with a rotating drum and the successive distributors fixed into the casing, removed.

Figure 5 shows a vertical section of a Parsons turbine, from which its principle of distribution can be easily understood. In the large turbines, the rotating drum carries several different diameters of bands of buckets. To equilibrate the longitudinal push or axial thrust, the drum is sometimes constructed symmetrically relatively to a medium plane perpendicular to the axis, but the number of rotating bands of buckets must be

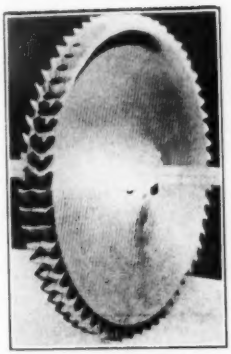


Fig. 1.—Rateau Helicoidal Wheel.

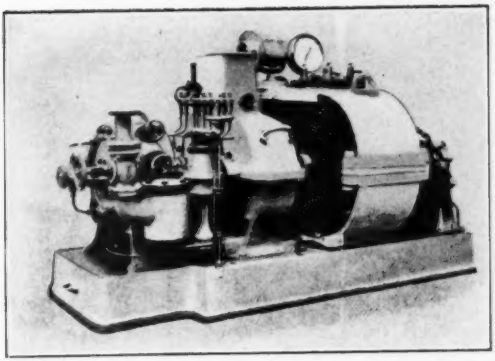


Fig. 2.—Turbine of One Wheel, Rateau System.

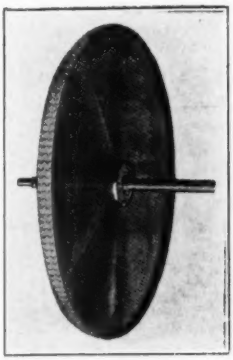


Fig. 3.—Riedler-Stumpf Wheel.

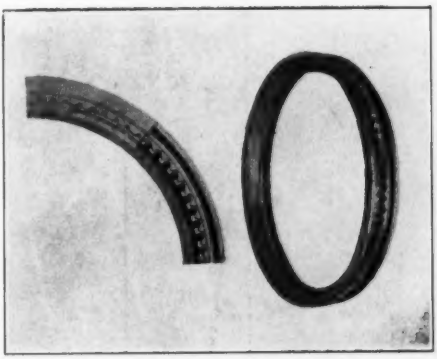


Fig. 4.—Riedler-Stumpf Distributors.

PLATE I.

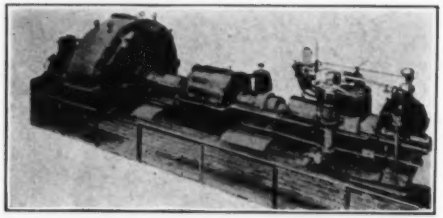


Fig. 5.—Parsons Turine, Elberfeld,

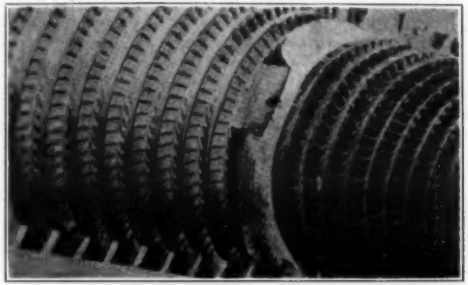


Fig. 7.—Interior View of Parsons Turbine.

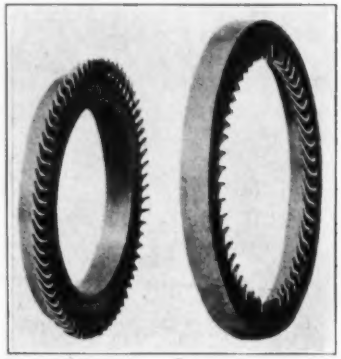


Fig. 6.—Rotating Band of Buckets and Distributor, Parsons.

Fig. 8.—Three-Wheel, Curtis.

PLATE II.

doubled in that case. For all the wheels the distribution is total, that is to say, the rotating bands of buckets are fed with steam over their entire periphery, which is a necessity when the steam acts reactively, as will be shown.

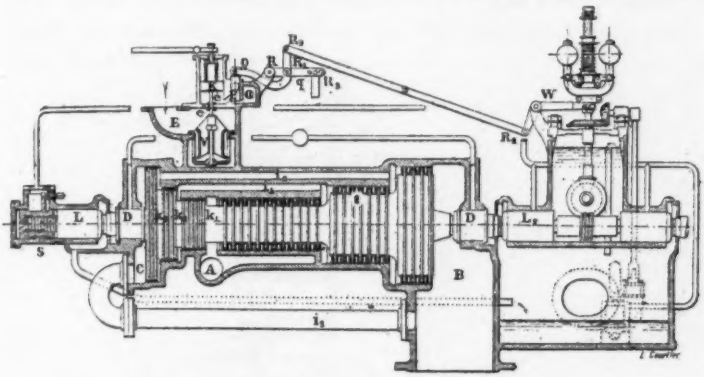


Fig. 5.—Vertical Section of a Parsons Turbine.

TURBINES WITH GROUPS OF WHEELS.

Instead of expanding the steam successively in each rotating wheel or moving band of buckets, the expansion can be effected by groups of wheels, each group using one expansion, or one fall or chute of steam, the speed of efflux created by the fall being maximum on the first wheel of the group while the succeeding wheels of the group only utilize the speed remaining at the outflow from this first wheel.

This principle of distribution was designed for the first time, it is believed, by Mr. Mortier under the name of distribution by stages of speed, as opposed to the distribution by stages of pressure as in the turbines of Parsons and of Rateau. This system could also be termed the turbine system with falls of speed.

Among the machines actually based on this principle, and which have already been constructed, is the American Curtis turbine. Some examples of this turbine will be given in order that its mode of construction may be understood.

Figure 6 represents a combination of a Curtis turbine with an alternator. It has two groups of three wheels each and is of 500 kilowatts power.

The ensemble of the Curtis turbine composed of three rotating wheels, is shown in Figure 8, Plate II.

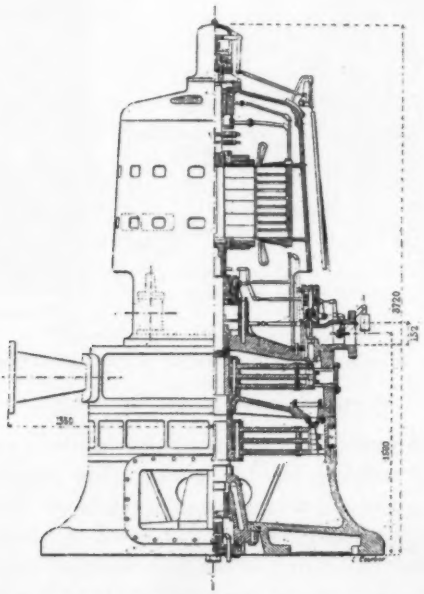


Fig. 6.—CURTIS ELECTRIC GENERATING SET WITH TWO GROUPS OF THREE WHEELS.

As will be seen, the first wheel utilizes the complete expansion of the steam, so that there results by means of the first distributor the full fall of the pressure of the steam admitted.

Between the first and the second wheel is a second distributor whose only effect is to utilize the speed remaining in the steam when it leaves the first wheel, and to thus send that steam into the second wheel with the proper direction.

A third distributor is placed on the exhaust side of the second wheel to utilize the speed there remaining in the steam,

but much less than at the exhaust of the first wheel; it directs the flow of steam to the third wheel.

Figure 7 represents one of the latest groups of electrogens and Curtis turbines; it is described in the issue of "Engineering" of the 5th February, 1904, Figure 2. The two groups of three wheels each, are replaced by four groups of only two wheels each.

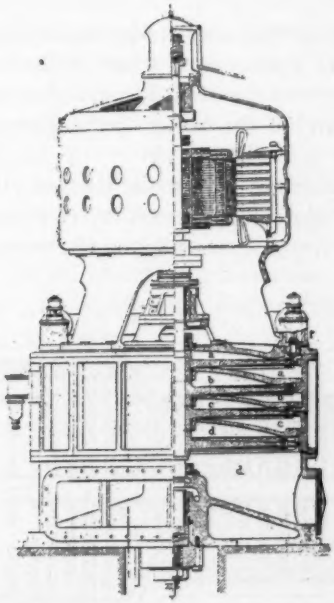


Fig. 7.—CURTIS ELECTRIC GENERATING SET WITH FOUR GROUPS OF TWO WHEELS.

According to the article referred to, this ensemble ought to furnish a power of 2,500 kilowatts at the speed of 750 revolutions per minute.

Observe that the Curtis turbine utilizing the steam by impulsion, as in the case of the Rateau turbine, will be identical with the latter when the number of wheels per group is reduced to unity.

In Figure 9, Plate III, are represented parts of the bucketing of a Curtis wheel for a power of 1,500 kilowatts. The buckets are dovetailed into the wheel and furnished with a tenon supporting a felly at the periphery.

MULTICELLULAR TURBINES.

The sole multicellular turbine actually in use is that of Rateau.

Its essential principle consists in employing the steam in different pressure stages; each wheel utilizes a certain determined fall of pressure, which is a fraction of the total fall from the pressure in the boiler to the pressure in the condenser.

Each wheel works in an isolated manner; it is enclosed in a small cell or apartment formed by two fixed diaphragms extending from the shaft to and into the casing.

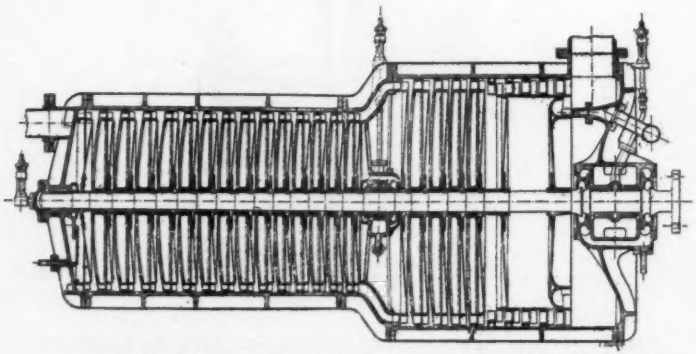


Fig. 8.—VERTICAL SECTION OF RATEAU TURBINE.

The vertical section, Figure 8, of a Rateau turbine, shows clearly this arrangement. The figure represents an old design with inside pillow-blocks, and was calculated for 1,200 horsepowers at 1,800 revolutions per minute.

The wheels used in the Rateau turbine are of plate iron.

At their periphery, fixed upon the felly, are the buckets provided for each band.

One of these wheels is represented in Figure 10, Plate III.

The distributors of the Rateau wheels are formed of curved buckets held between cheek pieces having the same curvature.

The distribution is made according to the principle of partial injection of steam, up to when the volume of steam required becomes such that there is necessity to inject it around the entire periphery of the wheel; in this latter case, the distributor occupies no longer an arc of the circle or divers arcs of the circle, but the complete circumference of the circle.

The distributors are fixed in logements or chambers arranged on the periphery of those discs which form diaphragms, and which constitute the front and the back of each of the wheels.

The diaphragms are constructed like a wheel with hub, spokes, and felly, covered over with an exactly fitting plate.

Figure 11, Plate III, shows a series of Rateau diaphragms with distributors to the periphery. These diaphragms are in a single piece and are of different diameters.

There can be employed either diaphragms in one piece, or diaphragms in two pieces.

The turbine whose section is given in Figure 8, is constructed with diaphragms in two pieces.

A view of that turbine, which has been the model chosen by the French Marine for the torpedo boat *No. 243*, is given in Figure 12, Plate III.

There will be remarked the dismounting facilities of this kind of construction.

Figure 13, Plate III, represents an ensemble of wheels and of distributors of a multicellular turbine with diaphragm in a single piece. On the shaft are keyed the different wheels and between them are placed the distributing diaphragms.

It is important to know that the clearance which could cause leakage of steam, leads down to the periphery of the shaft and into the annular space left free by the clearance for the hub of the diaphragm. This clearance of about 0.04 inch corresponds to a minimum space, and the proportion of steam which could escape from one wheel to the other without being utilized does not exceed in practice some hundredths.

There should also be remarked the holes pierced in the plate iron to avoid all longitudinal reaction.

Each wheel then moves in a cellule wherein the pressure is equal on all the surfaces.

Figure 14, Plate IV, represents a turbine of one body of 500 horsepowers measured on the shaft. The arrangement of the pillow-blocks should be remarked: they are outside of the body of the turbine, as well as the command of the regulator of the speed, which latter is supported on the principal pillow-block.

The body of this turbine contains two series of wheels of different diameters.

In Figure 15, Plate IV, is seen a turbine of two distinct bodies of 700 horsepowers on one shaft. It is necessary when the power surpasses a certain limit to fraction or divide the body of the turbine: there are experienced, in fact, great difficulties in keying on the same shaft too great a number of rotating wheels because the distance between the pillow-blocks becomes too large.

Figure 16, Plate IV, represents a marine turbine of two bodies, placed on two different shafts and combined in one connected series. The small body is termed "high pressure," and the other is termed "low pressure."

This turbine of about 2,000 horsepowers has been recently applied to the propulsion of a British torpedo boat.

Figure 17, Plate IV, shows a mixed type, high and low pressure and two bodies.

The large body called the "low pressure" can be separately supplied with the exhaust steam of a pumping engine pumping out a mine.

The "high pressure" body is supplied by two sources of steam, the one of 170 pounds pressure per square inch, the other of 70 pounds pressure per square inch, coming from different boilers.

As long as the exhaust steam is in sufficient quantity to do the full work of the machine, the "high pressure" body does not receive steam and an automatic connection puts it in com-

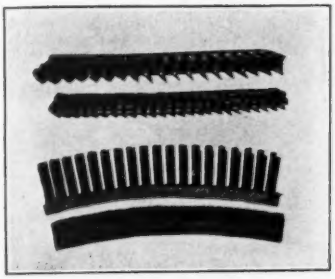


Fig. 9.—Parts of Bucketing, Curtis Turbine.

Fig. 10.—Rateau Wheel of Sheet Iron.

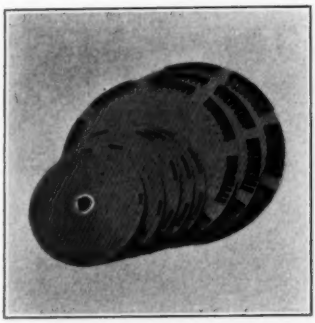


Fig. 11.—Series of Rateau Diaphragms.

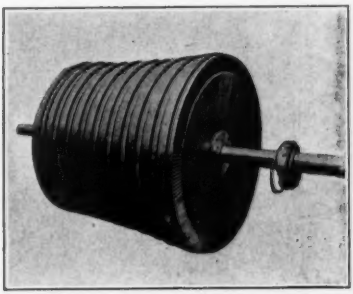


Fig. 13.—Ensemble of Wheels and Distributors, Rateau.

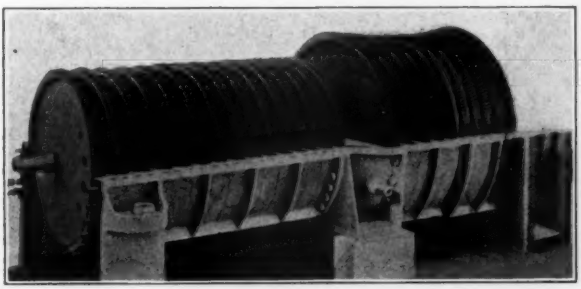


Fig. 12.—View of Turbine of Torpedo Boat 243.

PLATE III.

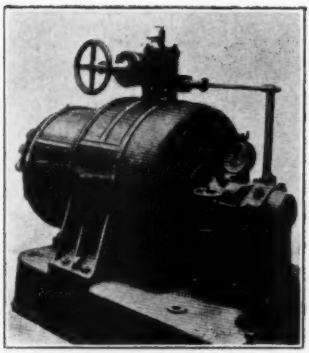


Fig. 14.—Turbine of one Body, 500 Horsepowers.

Fig. 16.—Marine Turbine of Two Bodies.

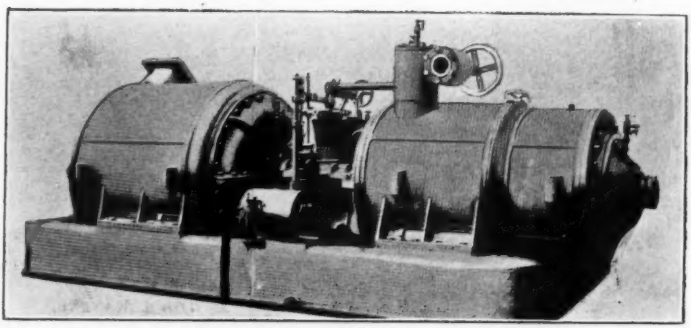


Fig. 15.—Turbine of Two Bodies, 700 Horsepowers.

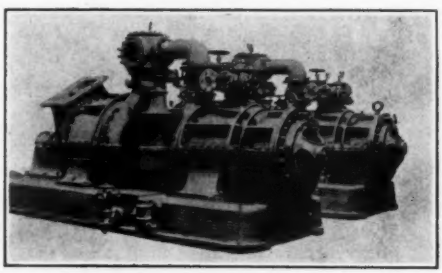


Fig. 17.—Mixed Type, High and Low Pressure.

PLATE IV.

munication with the condenser, so as to reduce the friction of the wheels on the steam to a minimum.

When the exhaust steam no longer suffices for doing the work, the arrival of steam takes place in the "high pressure" body, and the steam can thus expand in the ensemble of the two bodies.

There can consequently be utilized by this arrangement steam produced by or coming from, three different sources. It is superfluous to insist on the advantages of such a combination which appears to be quite unrealizable in a piston-engine.

II.—CLASSIFICATION ACCORDING TO THE ROUTE FOLLOWED BY THE FLUID-MOTOR.

The route followed by the Steam in passing through the Turbine, may be in either of two directions.

The steam can follow a trajectory parallel to the axis, or a trajectory normal to the axis.

In the first case the turbine is termed "axial" or "helicoidal"; in the second case it is termed radial.

AXIAL OR HELICOIDAL TURBINES.

Up to the present time, the only turbines employed in practice are of the axial variety.

It will suffice to pass only them in review.

In the Parsons turbine the fluid moves parallel to the axis. It enters either at one end of the turbine to flow out of the other end; or it enters into the diametral plan of the turbine and flows towards the two extremities.

In the Rateau turbine, the fluid follows a route also parallel to the axis; it enters at one extremity and escapes at the other.

In the Laval turbine, the steam enters by one of the faces of the wheel and leaves by the other. It is the same as in the first Rateau turbine, or in the Riedler-Stumpf turbine, the trajectory having the well known form of that of the Pelton wheel; the fluid enters by the diametral plane of the wheel, and leaves by the two faces of the wheel, but its direction is always parallel to the axis of the wheel. The Curtis and Westinghouse turbines are likewise axial turbines.

RADIAL TURBINES.

The Radial Turbines are divided into two groups:

Centrifugal Radial Turbines,

Centripetal Radial Turbines.

In the first, the fluid flows from the axis of the wheel to its periphery.

In the second group, the fluid, on the contrary, enters at the periphery and leaves at the axis of the wheel.

CENTRIFUGAL RADIAL TURBINES.

There can be cited as the first, the Centrifugal Radial Turbine of Dow, composed of a series of rotating concentric circlets of buckets into which the fluid is delivered successively from a corresponding series of directrices.

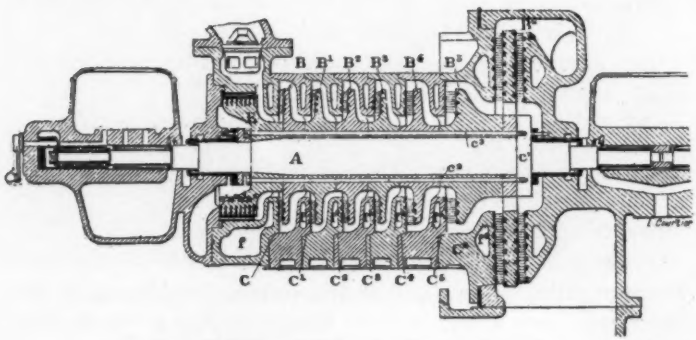


Fig. 9.—PARSONS' CENTRIFUGAL TURBINE.

The Multicellular Centrifugal Radial Turbine of Edwards, in which the steam traverses the distributing canals or passages directed from the axis to the periphery, and acts on rotating buckets fixed to the body of the wheel.

At the extremity of the turbine, the expanded steam acts in an inverse manner to the centripetal flow.

Parsons has also exercised his ingenuity on the construction of a centrifugal radial turbine represented in Figure 9.

The construction is somewhat analogous to that of Dow's

turbine. The rotating circlets of buckets are concentric, and at the periphery of each series the flow of steam is led into the axis of the following series by an oblique passage or canal.

CENTRIPETAL RADIAL TURBINES.

One of Parsons' patents shows a Centripetal Radial Turbine (Figure 10); the construction is the inverse of the preceding turbines.

Dumoulin, also, has invented a Centripetal Turbine of which a description is given in Sosnowski's work.*

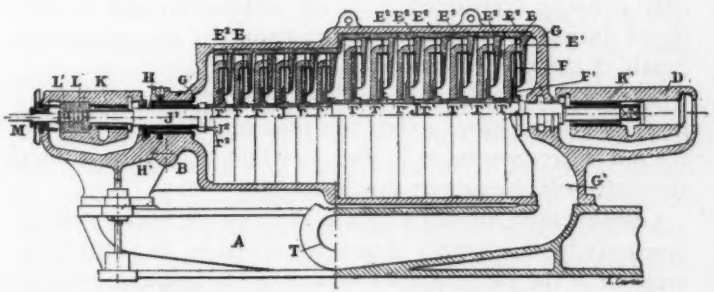


Fig. 10.—PARSONS' CENTRIPETAL TURBINE.

Finally, in the same Memoire will be found the description of Last's original machine, in which each wheel works centrifugally and centripetally; the flow of the steam is always slightly oblique to the axis; so that this machine is simultaneously a heliocentrifugal and a heliocentripetal turbine.

III.—CLASSIFICATION ACCORDING TO THE MODE IN WHICH THE MOTOR-FLUID FUNCTIONS.

The steam can act in the Turbine according to two different Modes.

It can function by action or impulsion, and by reaction.

The steam, in its mode of action, completely expands in the distributor, the fall of pressure being wholly employed to create all the speed of which the steam is susceptible; the

*Roues et Turbines à Vapeur. Paris. 1901.

energy derived from the expansion is found integrally in the *vis viva* of the molecules of the steam thus put in movement.

The push on the movable buckets is then wholly due to the quantity of movement of the steam, this push being rigorously equal to the product of the mass into the speed, taking into account, in the calculation, the effect of the variable inclination of the steam on the rotating buckets during its action; the density remaining invariable during the route across the buckets, the cross section of the route can remain constant.

In the method of reaction, the expansion of the steam is only partially accomplished in the distributor, and is completed during the trajectory of the steam in the passages or canals of the moving buckets.

The speed created by the outflow from the distributor, for a given fall of pressure, is then less than in the preceding case, the fall of pressure between the entry into, and the exit from, the distributor being very much less.

Consequently, the steam acts in the rotating buckets simultaneously by its quantity of movement and by its pressure, in measure as the expansion takes place in the rotating buckets.

The speed of circulation increases, and the cross section of the passage ought not to remain constant. When the steam does work according to this mode of action, the two sides or faces of the rotating wheel are subjected to the same pressure, since it rotates in an entirely expanded fluid.

On the contrary, in the reaction turbine, the two sides of of the rotating wheel are subjected to different pressures, and this difference is so much the greater as the degree of reaction is, itself, greater.

LAW OF EFFLUX OF SATURATED STEAM.

The mode of functioning of the fluid-motor depends essentially on the form and on the direction of the buckets, passages, or channels, in which it moves, be they fixed or rotating.

The form of these buckets, passages, or channels, is determined by the law of efflux of saturated steam.

The efflux of elastic fluids by means of rectangular or curved

distributors, differs much from the efflux of liquids for the cases in which the proportion of the higher pressure to the lower pressure producing the flow is notably less than unity.

Calling I the weight of the discharge per second, S the area of the cross section of the stream of discharge, δ the specific weight of the fluid in that cross section, and V the velocity per second of the discharge, these quantities have to each other the simple relation

$$I = SV\delta,$$

which is the equation of continuity.

If the specific weight δ be constant, as in the case of liquids, S is inversely proportional to V, and as the velocity V increases always in the same measure that the pressure decreases, S continuously diminishes. Thus, the distributing canals of hydraulic turbines are always made convergent. With elastic fluids, this is not the case; with steam, the specific weight decreases in measure as the pressure decreases during the increase of velocity, in such a manner that the product, which at first was continuously increasing, passes by a maximum and becomes continuously decreasing.

In the case of a gas, the maximum occurs for the proportion

$$\frac{\eta}{P} = 0.52;$$

with steam, this proportion is about 0.58, whence results that when the lower pressure η is lower than 0.58 of P, the tuyere of efflux, at first converging, should then become diverging, if the steam is to be expanded still more, in order that it may obtain all the velocity possible for its fall of pressure from P to η. The proportion of the area of the final cross section of the tuyere to the cross section at the section S of the neck, should vary with the proportion $\frac{\eta}{P}$ of the pressures, while the pressure at the neck is always equal to 0.58 of P.

The quantity of steam discharged is independent of the low pressure η when that pressure is less than 0.58 of the high pressure P; but, on the contrary, the quantity of steam dis-

charged depends also on the pressure η when that pressure is greater than 0.58 of the pressure P.

There are then two very different cases to be distinguished:

In the first case, the calculation of the quantity of discharge depends only on P, and the formula is simple. It can be expressed as follows, as shown by Mr. Rateau:

$$I = S\,P\,(15.20 - 0.96\ log\ P).$$

In the second case, the calculation of the quantity of discharge depends simultaneously on P and on η.

In both cases, the smallest section of the tuyere (at the neck, if the tuyere be convergent-divergent, or, at the discharging end, if it be only convergent) must be taken for the calculation of the quantity of discharge.

Thus, when the lower pressure η is greater than 0.58 of P, which is the general case of steam-turbines with multiple wheels, convergent tuyeres must be employed.

On the contrary, when η is smaller than 0.58 of P, which is the case of steam-turbines with one wheel, like Laval's turbine, convergent-divergent tuyeres must be employed if the complete expansion of the steam in the tuyere be desired in order that the expanded steam may have all the velocity which corresponds to its fall of pressure from P to η.

Figure 11 shows the curve giving the values of the successive diameters of a conical tuyere in function of proportion of expansion. The abscissae show the ratio of expansion of the lower pressure η to the higher pressure P, and the corresponding ordinates show the values of the diameters.

When the ratio of the lower pressure to the higher pressure falls below 0.58, the tuyere becomes convergent-divergent; the divergent part has unfortunately an injurious influence on the economy. As Mr. Rateau has already observed in his turbines, and as Professor Stodola has experimentally demonstrated, the stream of steam has a tendency to separate from, or to leave the surface of, the tuyere in the divergent part, thus producing a back pressure against the steam-current at the moment of this separation.

To avoid, at least partially, this defect, an angle of small conicity, not surpassing 3 or 4 degrees, must be given to the divergent part of the tuyere. Under these conditions, when the ratio of the expansion is considerable, the convergent-divergent tuyere should have a great length, which can be particularly inconvenient when the discharge of steam is considerable.

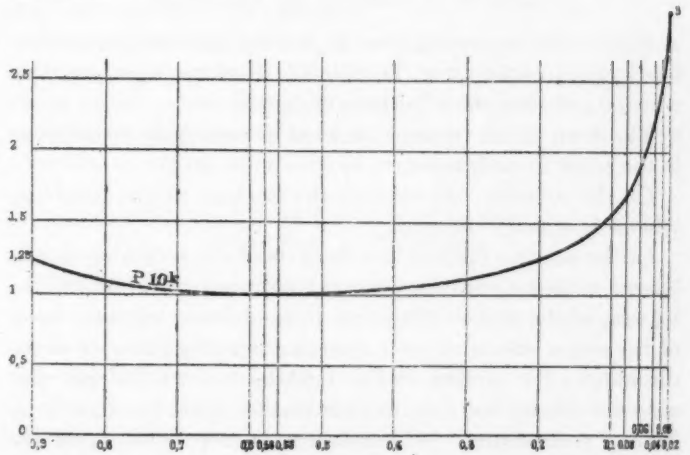


Fig. 11.—PROFILE OF THE TUYERES FOR A STEAM TURBINE.

Supposing a minimum section of unit diameter, and a discharging section of treble this diameter, then, that the total angle of divergence remain at 4 degrees, the length of the tuyere must be equal to 29 times the diameter of the minimum section.

With a tuyere discharging, say 441 pounds weight of steam per hour, and with a pressure P of 147¼ pounds per square inch above the absolute zero, the minimum section would have a diameter of 0.028 inch, the length of the divergent will be 0.812 inch, the expansion being only 2 per centum, which corresponds to the lower pressure η of 2.94 pounds per square inch.

The necessity of giving a great length to the converging-

diverging tubes, compels their number to be increased in order to reduce their diameter, and, in fact, all their dimensions. It is a serious matter with steam-turbines of one wheel, but it does not exist for steam-turbines of multiple wheels, in which the tuyeres are all convergent.

COMPARISON OF THE BUCKETING OF A TURBINE OF ACTION AND OF A TURBINE OF REACTION.

Figure 12 represents the Bucketing and Steam-passages, fixed and mobile, of a Turbine of Reaction with multiple wheels, and those of a Turbine of Action.

The form of the passages in fixed or stationary distributors is the same in both cases.

On the contrary, the form of the Buckets in the revolving wheels is notably different.

In the reaction turbine, the Buckets of the revolving wheels have a section decreasing from the entrance of the fluid to the leaving of the same. This fact is due to the expansion itself of the steam whose velocity increases from the entrance to the discharge; the section of the buckets follows the law just shown. To realize this law, the bucket must be slightly increased at its center. All experiments have shown that the mobile buckets can be constructed of plate iron of equal thickness and of circular form placed symmetrically to the diametrical plane of the axis: the section of the bucket is not then rigorously constant, but the difference from the theoretical section is slight and the economy is not sensibly affected.

To distinguish, at a glance, the wheel of a turbine of reaction from the wheel of a turbine of action is easy.

In the first, the buckets are not symmetrical in relation to the diametrical plane; and, in the second, they are symmetrical.

The same figure, 12, gives the diagram showing the velocities in a turbine of action.

The absolute efflux of the steam at the entrance, is given with the angle of inclination of the distributor.

This speed is compounded with two other speeds:

The speed of entrainment of the steam, or peripheral speed of the wheel, and the relative speed of the steam in the buckets; this is inclined from the angle itself of the bucket at the entrance.

The relative speed of the steam at its discharge is naturally less than its relative speed when entering. This reduction is due to frictions and eddies of the gaseous steam during its trajectory through the passages.

By compounding this last relative speed with the speed of entrainment, there will be obtained the absolute speed of the steam at the moment of its discharge, and its direction.

From the diagram of the speeds, the hydraulic economy of a turbine of action can easily be calculated.

CONSEQUENCES OF THE MODE IN WHICH THE FLUID-MOTOR FUNCTIONS IN TURBINES.

The mode in which the steam functions, whether by impulsion or by reaction, has very important consequences for these two categories of turbines.

In the turbine of reaction, it is necessary:

1st. To equilibrate the axial push or thrust produced by the difference in the pressure of the steam upon the two faces of each revolving wheel.

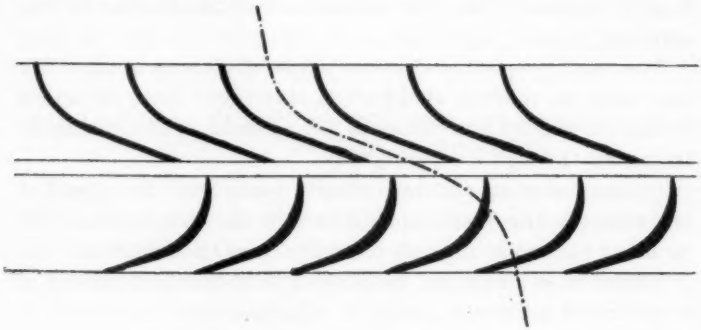
2d. To allow but a very slight clearance between the stationary and the moving parts in order to prevent steam-leakage from one face to the other of the wheels.

3d. To admit steam simultaneously to the entire periphery of the wheel, in order to avoid loss by eddies when the canals of the moving wheels arrive opposite to those of the stationary wheels.

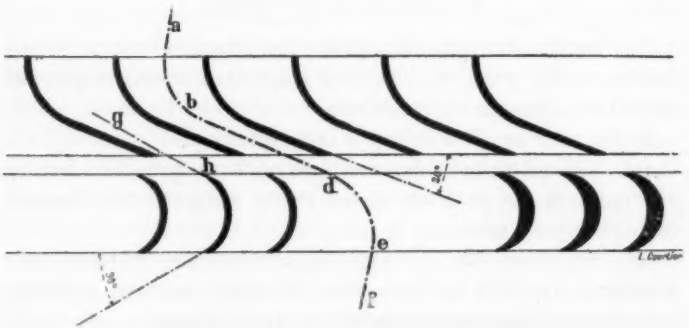
The schematic design in Figure 13, shows the fundamental differences which separate the functioning of a drum turbine of reaction in the Parsons system, from that of a multicellular turbine of action, both turbines being helicoidal.

The mode of action causes a strong difference of pressure on the two faces of the bands carrying the buckets; and, further, the total difference of pressure between the extreme

pressures P and γ, acts on the two exterior faces of the drum carrying the buckets.



TURBINE OF REACTION WITH MULTIPLE WHEELS.



TURBINE OF ACTION WITH MULTIPLE WHEELS.

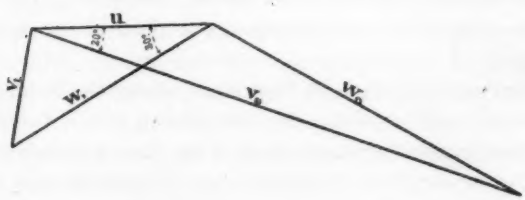
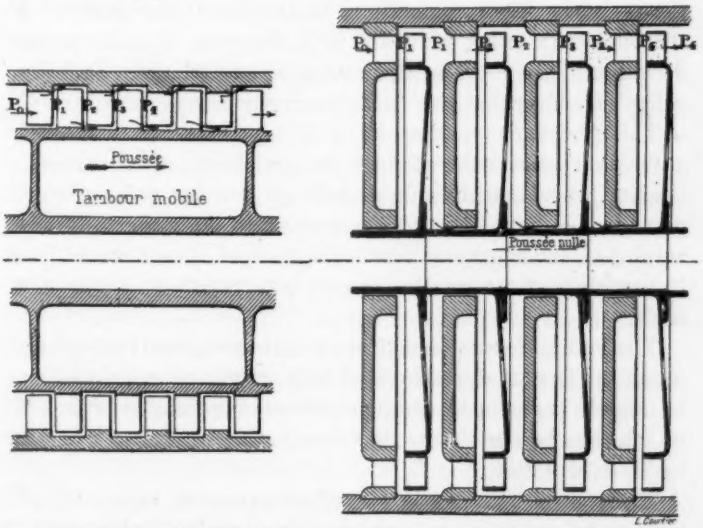


DIAGRAM OF SPEED FOR A TURBINE OF ACTION.

Fig. 12.

This axial pressure, which, for large turbines, may amount to several thousands of pounds, has to be equilibrated.

In the Parsons system, this pressure is equilibrated by making the steam pass simultaneously in two opposite trajectories; or, by adding an equilibrium cylinder with movable piston to the extremity of the turbine, steam being admitted to the two sides of the piston and respectively of the two pressures P and η.



TURBINE OF REACTION WITH MULTIPLE WHEELS, PARSONS SYSTEM.

MULTICELLULAR TURBINE OF ACTION, RATEAU SYSTEM.

Fig. 13.—PATH OF STEAM AND AXIAL THRUST IN HELICOIDAL TURBINES.

In a multicellular turbine of action there is no unbalanced axial pressure to be counterbalanced.

In the reaction turbine, the clearance between the rotating wheels and the envelope casing, or shell, should be reduced to a minimum. And the same may be said of the clearance between the exterior periphery of the drum and the internal face of the fixed band of distributors.

In the multicellular turbine, the clearance can be consider-

able, and attains, in practice, as much as from 0.12 to 0.16 inch between the fixed part, distributor or envelope, and the rotating wheel. Only the clearance between the shaft and the internal hub of the distributor should be reduced to about 0.04 inch.

When the movable buckets of a reaction turbine, full of steam, come by its movement in front of a fixed or blank side, which will happen if the distributor extends over only an arc of the periphery, there is produced by the fact of the pressure in the buckets a reflex of steam in a direction opposite to the direction of the movement, causing strong frictions and considerable eddies that absorb a part of the energy.

This phenomenon compels, in the case of reaction turbines, a distribution over the whole of the periphery; also it compels the first wheels on the side of the high pressure to be of small diameter, because, for high-pressure steam, if large wheels be employed, the aggregate sections of the distributor would lead to partitions of too small a height, not exceeding 0.04 to 0.08 inch.

The hydraulic economy of small diameter wheels being very much less than that of wheels of large diameter, it is necessary to notably augment in reaction turbines the number of wheels which thus becomes more than double the number required in turbines of action.

The remark may not be superfluous that the mode of functioning by action when employed in a multicellular turbine has very serious advantages, both as regards the construction and as regards the simplification of the machine.

REMARKS ON THE CLASSIFICATION OF STEAM-TURBINES.

The conclusion that clearly results from the general classification of steam-turbines, and which the examination of the economy given by them confirms, is that all the various systems designed merge in two forms, namely, the drum reaction turbine with multiple bands of bucketing, devised by Parsons; and the multicellular turbine of action devised by Rateau.

By combining two or several of the systems of each category,

new systems can be formed participating in the properties of the different systems selected. In this manner, for example, a drum turbine of reaction can be combined with a multicellular turbine of action.

And, still further, by taking on the one part the wheels of the Pelton system with tuyere distributors as first employed by Rateau and later by Stumpf; and, on the other part, by taking the helicoidal wheels as afterwards employed by Rateau; to constitute a multicellular turbine of action of a special system.

But all these hybrid systems reunite the disadvantages of their different prototypes composing them, without having all of their advantages.

Although the drum turbine of action with bands of moving buckets, compared with the drum turbine of reaction, permits, like all turbines of action, partial injection of the steam and a larger clearance between the buckets and their casing or envelope, it restricts, on the contrary, to a smaller clearance between the drum and the internal periphery of the distribution for preventing the internal leakage of steam.

These leaks are greater than in the Parsons turbine, because the expansion being complete in each distributor, the difference of pressure between the two faces will be a maximum.

In the Rateau multicellular turbine, the clearance in question is carried down to the periphery of the shaft, and the leakage is proportionally reduced.

The drum turbine of action participates, it is true, in the property of the turbines of action to avoid all axial thrust on the bands of moving buckets, but they have that thrust on the drum itself, the faces of which are subjected to the difference of pressure between the maximum and the minimum pressure in the turbine.

It will then be perceived that this type, though based on the principle of action, does not benefit by all of the advantages of that principle, and it retains several disadvantages of the principle of reaction.

The mixed multicellular turbine, having Pelton wheels and

helicoidal wheels, has the disadvantage of a mediocre economy for the wheels of the first kind under penalty of either multiplying their number and of rendering the construction costly, or of too much increasing their diameter.

The velocity remaining in the steam when leaving the Pelton wheels is absolutely non-usable, which still more contributes to lessen the economy. Also, there can easily be foreseen that this mixed system will, by the suppression of the Pelton wheel, fatally end in the Rateau multicellular turbine, and that the preceding mixed system will end in the Parsons turbine.

The final experimental result, then, and to which the logic of physical and mechanical laws also leads, is that there can be only two fundamental forms of steam-turbines, namely, that of Parsons and that of Rateau, of which all others are but variants.

ECONOMY [RENDEMENT] OF STEAM-TURBINES.

DEFINITION OF THIS ECONOMY.

The economy of a steam-engine is one of the most universally accepted notions among engineers, and, at first view, seems one of the most precise.

In reality, the fact is very different, and, notwithstanding the works of numerous scientists and experimenters, nothing can be less definite than the economy of a steam-motor, taking the term at hazard as employed by any constructor whomsoever.

Even yet, for a great many persons, the economy is measured by the weight of fuel consumed by any given engine per unit of work done, the definition of which is often of the most vague kind.

This mode of measurement which involves in it the economy of the boiler and of the auxiliary apparatus, can be still more vitiated by the manner of considering what is coal; whether the ash, clinker and hygroscopic water that it may contain be or be not included in the term, and all of which is very vari-

able for different coals, as well as the calorific power of different coals, which is quite as variable.

As for the work done by the motor, it was for a long while measured by the dimensions of the cylinder which was supposed capable of producing a given power at a determined speed.

Later, the discovery was made that a more certain method was necessary, and then the weight of steam actually used was substituted for the weight of coal consumed.

When the problem concerns engines of great power, the determination of the weight of steam used is a much more delicate operation than it seems at first sight.

As regards the measurement of the useful power developed upon the shaft, it is always difficult when the power exceeds 40 or 50 horsepowers, unless the motor is directly attached to a generator of electricity.

The only process in general use—indicator diagrams taken from the cylinder—does not in most cases give the economy with sufficient precision.

The diagram gives well enough the internal or indicated work done, but as the proportion between the effective work really done on the shaft and the indicated work is always unknown unless the effective work can be directly measured, the information furnished by the indicator must necessarily be received with caution.

The proportion between the effective work and the indicated work, varies, according to authors, from 75 to 93 per centum: 90 per centum is generally admitted for a steam-engine in good condition, but a rigorous demonstration cannot be given, as that requires the use in each case of a dynamo performing the rôle of a dynamometer.

Only recently have the processes employed in electrical technology been applied to measure with precision the work done upon the shaft of a generator of electricity by its steam-motor.

This method of measuring is nearer exact than any of the other methods attempted; measuring by the brake being im-

possible when the power developed by the engine, and its rotary speed, surpasses certain limits.

Once the number of effective horsepowers developed by a steam-engine, and its corresponding consumption of steam, being known, what is intended by its economy [rendement] must be exactly defined.

The economy of a steam-turbine is the thermal economy represented by the relation between the effective energy furnished to the shaft and the total disposable energy contained in the corresponding weight of steam used per unit of time; the fall of pressure which renders this energy disposable, is equal to the difference between the pressures of the steam entering and leaving the machine.

To calculate the energy of a given machine, the effective power of which and its consumption of steam has been ascertained, the total energy disposable under the conditions of its use must be known.

CALCULATION OF THE TOTAL DISPOSABLE POWER CONTAINED IN THE STEAM.

The calculation of the total disposable power contained in a pound weight of steam expanding from its higher pressure P to its lower pressure η, is given by the laws of thermodynamics and by the experiments of Regnault.

This calculation can be made either by the aid of formulas, or by the aid of the entropic diagram the use of which is daily becoming more and more extended.

During the investigations of the steam-turbine made by the author in 1896, he constructed a table that gave for any fall of pressure, whatever, the disposable energy of one pound weight of steam.

This table having been calculated, was sent to Mr. Rateau who conceived the ingenious idea of translating it under the form of an abacus with a logarithmic base, reuniting on one line the items corresponding to an equal consumption of steam.

This abacus is represented in Figure 14; the abscissas are

the absolute high pressures P, and the ordinates are the absolute low pressures γ.

In constructing this abacus, Mr. Rateau soon perceived that the lines of equal consumption of steam were not curves as is generally the case with elastic fluids, but straight lines. The coincidence was so perfect that he was enabled to immediately deduce an empirical law for the calculation of the theoretical

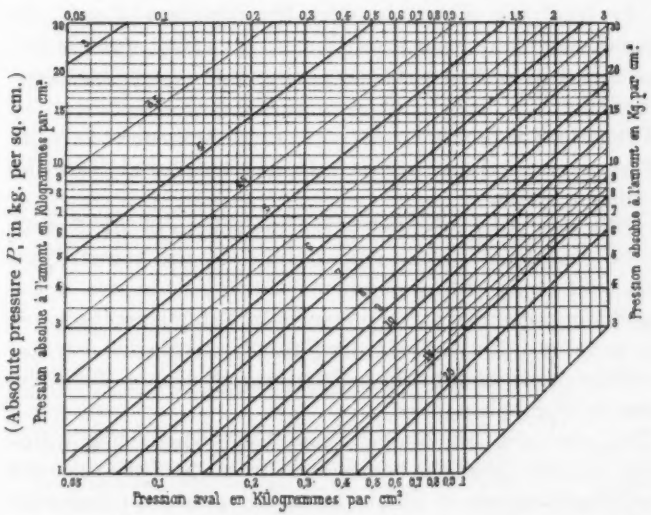

(Absolute pressure P, in kg. per sq. cm.)
Pression absolue à l'amont en Kilogrammes par cm²
Pression absolue à l'amont en Kg par cm²
Pression aval en Kilogrammes par cm²
(Absolute pressure γ, in kg. per sq. cm.)

Fig. 14.—Abacus of Theoretical Consumption for a Steam-Engine.

consumption of steam per horsepower per hour in pounds weight of steam evolved between two given pressures. This empirical law is remarkably in accord with the results drawn from the tables of Regnault; the differences are about the one thousandth part.*

Knowing, by means of Mr. Rateau's abacus, the theoretical consumption K represented by the effective horsepower per

*_Abacus of the theoretical consumptions by a steam-engine, and new laws relative to steam_, by Mr. Rateau. Extract from Annales des Mines, issue of February, 1897.

hour in the given conditions, the speed of outflow of the saturated steam between the same high and low pressures P and η can be easily deduced. The simplified formula is the following:

$$V_0 = 100 \sqrt{\frac{530}{K}},$$

which gives the speed V_0 of the outflow of the steam in meters [3.2809 feet] per second, between the pressures P and η in kilogrammes per square centimeter, [14.2228 pounds per square inch], K being the corresponding theoretical consumption.

The values of the theoretical consumption furnished by the abacus, are entirely in accord with those deduced from the entropic diagram.

In the case of superheated steam, the abacus of Mr. Rateau can be completed by calculation, aided by the thermic diagram; the superheating augments the utilizable area of the diagram between two given pressures by a supplementary area easy to value. Mr. LeLong, Engineer of the Marine, has furthermore shown an ingenious abacus which enables the calculation to be supplemented.

The abacus of Mr. Rateau, completed when superheated steam is used, allows the valuation to be made of the economy of all steam-engines of which the higher and lower pressures, P and η, are perfectly known and of which the consumption per effective horsepower per hour is known. A precise notion is thus obtained of the economy, which furnishes a truly rational classifying of a steam-engine whatever may be its nature and its mode of construction. The economy should no longer be subordinated to the consumption of steam.

ECONOMY AND CONSUMPTION OF STEAM.

The net economy, and the consumption of steam, are two quite distinct notions, though too often confounded: when the consumption of one machine is given in order to compare it with that of another, the higher and lower pressures P and η

are not always stated between which the steam is evolved. It can very easily happen that the consumption which appears most advantageous corresponds, in reality, to a lower economy.

The following table shows an example of two steam-turbines, one of which has a small consumption but with the steam received under a high pressure P and expanded down to a feeble pressure η; the other machine, on the contrary, has a large consumption, but then the steam is admitted to it at a notably less pressure, and its vacuum is mediocre.

The table shows that although the consumption of steam by turbine A is 24 per centum less than by turbine B, the economy of the latter is 21 per centum more.

Whence results that were turbine B placed in the conditions of turbine A, its consumption would be 18 per centum less.

TABLE OF THE COMPARATIVE CONSUMPTION OF STEAM BY, AND OF THE ECONOMY OF, TWO TURBINES A AND B UNDER DIFFERENT CONDITIONS.

	Turbine A.	Turbine B.
Higher pressure P, in pounds per square inch above zero	220.84	147.23
Lower pressure η, in pounds per square inch above zero	1.40	1.16
Superheating added, in degrees Fahrenheit	212.00	0.00
Temperature, in Fahr. degrees, of the steam	566.60	356.80
Consumption of steam, in pounds weight per hour, per effective horsepower	12.12	15.10
Consumption of steam, in pounds weight per hour, per theoretical horsepower	6.46	9.81
Thermic economy	$\frac{6.46}{12.12}=0.534$	$\frac{9.81}{15.10}=0.650$
	Consumption.	Economy.
Ratio of $\frac{B}{A}$	1.246 or 24.5 per ct. in excess.	1.217 or 21.5 per ct. in excess.

Turbine B in the conditions of Turbine A. Consumption of steam in pounds weight per hour per effective horsepower $6.46 \div 0.65 = 9.939$ pounds, is 18 per centum less.

HYDRAULIC ECONOMY OF THE WHEEL OF A TURBINE OF ACTION.

The most important term which intervenes in the total economy of a turbine of action, is that which is called the

hydraulic economy. It is calculated from the diagram of the speeds on entering and on leaving the moving bucketing. This method has been proposed by Mr. Rateau for the calculation of hydraulic turbines. It was applied by him for the first time to steam-turbines in a Memoir on the Elementary Theory of Steam-Turbines, deposited in the Academy of Sciences in 1901.

The calculation of the hydraulic economy depends essentially on the coefficient of the slowing of the relative speed of the fluid in the moving bucketing; this coefficient once known as well as the absolute speed on entering, and the angles of the fixed and moving bucketings, the hydraulic economy can be immediately deduced.

The calculation can be made graphically, as shown by Mr. Rateau, as it is easily translatable into formula.

The hydraulic economy for the given angles of construction of the fixed and moving bucketings, and for a known coefficient of the slowing of the speed, is expressed, in function of the relation of the peripheral speed of the bucketing to the absolute speed of the steam at its entrance; this speed itself being given, the economy is then known in function of the peripheral speed of the wheel.

Both theory and experiment show that the hydraulic economy increases rapidly with the peripheral speed up to a maximum after which it gradually decreases.

In general, the speed corresponding to the hydraulic economic maximum cannot possibly be attained with turbines of multiple wheels, but it can be approximated sufficiently near to realize hydraulic economy up to 70 per centum of the maximum and even surpassing that proportion.

STEAM-LEAKS IN TURBINES OF MULTIPLE WHEELS.

Steam-leakage does not exist in turbines of impulsion with one wheel, because the steam acts only in a single compartment.

In turbines of multicellular action, leakage is only to be feared between the two faces of a distributor; the steam which

escapes from one face to the other, loses thus its pressure of expansion without any utilization.

There has been previously shown that in the Rateau turbine this defect is almost completely corrected.

In the reaction turbines with multiple wheels, the leakage at the distributor is less, but the leakage around the moving bucketing is considerable; the steam has always a tendency to pass from the one face to the other, since the pressures on these faces are different. The tendency to this leakage is nearly the same from one end of the turbine to the other, while in the multicellular system of action, the leakage around the shaft has a sensible value for only the first wheels where the difference of pressure between the two faces of the distributor is more considerable.

This is another reason that compels increasing the number of wheels in the reaction turbine.

FRICTION OF THE STEAM.

The hydraulic economy is never equal to unity, as has been previously remarked: it tends toward a maximum which it cannot reach in practice.

The difference between the value of the hydraulic economy and unity, constitutes the most important loss of energy in the steam-turbine. This loss of energy is due, on the one part, to the incomplete utilization of the speed of the steam, and, on the other part, to the frictions and eddies of the stream of steam in its passage across the moving bucketing.

A second loss, the importance of which is not negligible, is caused by the frictions of the wheel and of the bucketing on the steam in the case of one or several wheels. These frictions are equally important for drum turbines where the moving bands of buckets, and the cylindrical surface of the drum rub, each in its turn, upon the steam.

In the case of a moving wheel, the work lost by its friction is proportional to the density of the fluid, to the cube of the angular speed, and to the fifth power of the diameter. As the

density of saturated steam is sensibly proportional to the absolute pressure, then for equal diameter, the frictions are considerable for the first wheels which receive the steam at high pressure, and nearly negligible for the turbines that revolve in a medium having a pressure but little above the pressure in the condenser.

Thus, the turbine-wheel receiving steam of the pressure 147.23 pounds per square inch absolute, has a loss of work by friction a hundred times greater than the same wheel revolving in steam of the condenser-pressure of 1.55 pounds per square inch absolute.

The first wheel of a turbine must consequently have a diameter very much less than its last wheel; the different diameters being calculated in such a manner as to sensibly equalize the loss by friction.

An analysis of this friction of the wheels, reveals the cause of the excellent economy of low pressure turbines utilizing exhaust steam at atmospheric pressure according to the ingenious system invented by Mr. Rateau, while, with piston steam-engines, it is the low pressure cylinder that gives the poorest economy, either by the increase of its frictions or by its thermic losses. On the contrary, in the steam-turbine, it is the low pressure wheels that give the maximum economy.

The importance of the friction of the wheels, has often been objected to in the multicellular system. The experiments made for determining the law of the frictions, and the results obtained with machines calculated according to the coefficients furnished by practice, have justified the indications of theory that machines of this nature can be constructed in which the frictions will be represented by only from 2 to 4 per centum of the total work. These figures are comparable with those that have been obtained with drum turbines, in which account must not only be taken of the frictions of the moving buckets, but also of the frictions of the cylindrical surface of the drums.

LOSSES BY EDDIES.

The losses by eddies can be considerable, unless precautions are taken to limit them.

In the reaction turbines, as has previously been stated, the fluid which gradually expands in the moving buckets has naturally a tendency to return backward to the orifices by which it entered these buckets. To prevent this effect, these buckets must be kept constantly filled with gaseous steam flowing in the same direction that they move. Now this result can only be obtained by feeding the buckets in a continuous manner. Consequently, the reaction turbine requires, as has previously been said, feeding around the entire periphery of its wheels.

Experiment has shown that if partial injection of steam be applied to a reaction turbine, the economy will be very considerably lowered. Partial injection is possible, then, only for turbines of action.

In the turbines of action with one wheel, where the total fall of pressure is utilized by a single distributor, the form of the tuyere is necessarily convergent-divergent; unfortunately, the divergent part is always an organ of bad economy. Phenomena of the adhesion of the steam to the inner surface of the tuyere, causes losses of economy which the author has often measured, and which attains in the best conditions from 10 to 12 per centum of the total energy contained in the steam. This constitutes a grave inferiority for turbines of one wheel, whatever may be their mode of construction, unless their employment be restricted to minimum falls of pressure, the absolute high pressure not exceeding double the absolute low pressure.

Another cause of eddies exists in the trajectory of the steam leaving the moving bucketing.

If the turbine be so accurately constructed that the steam escaping from it exactly meets the openings in the following distributor, the principal part of the speed remaining in the steam current will be utilized. This remaining energy is not

an important part of the total energy, but in multicellular turbines it reaches 5 or 6 per centum.

On the contrary, if the steam current leaving the turbine strike an opposing surface, its remaining speed will be entirely lost, and the corresponding energy will be wasted in concussions and eddies.

In multicellular turbines using the pressure in successive steps or stages, it is indispensable to so exactly fit the successive distributors that the remaining speeds of the steam will be utilized.

CALCULATION OF THE ECONOMY.

For a given wheel and a given fall of pressure, that is to say, for a speed of efflux of the steam determined in advance, the hydraulic economy varies with the peripheral speed of the wheel. A curve can then be traced of the hydraulic economy in function of the peripheral speed, which permits the ascertainment in each case, of the value that can possibly be attained in practice.

After the hydraulic economy has been determined, there must be deducted from it the losses by leakage, by friction, and by eddies.

Take, for example, a Rateau multicellular turbine furnishing 1,500 effective horsepowers to the shaft at the speed of 1,500 revolutions per minute.

By the aid of the practical coefficients obtained from experiment, the hydraulic economy of such a machine can easily be calculated; it is equal to 0.69. The losses by steam leakage and by the frictions of the pillow-blocks, absorbs $1\frac{1}{2}$ per centum of the power. The losses due to the frictions of the wheels upon the steam are $2\frac{1}{2}$ per centum, making a total of 4 per centum.

The net economy on the shaft at the speed indicated, will then be $0.69 \times 0.96 = 0.662$.

The following table shows the comparative elements of the economy of the same turbine at speeds of 1,500 and of 2,000 revolutions per minute.

CALCULATIONS OF THE ECONOMY OF A RATEAU MULTICELLULAR TURBINE OF 1,500 HORSEPOWERS.

——	At 1,500 revolutions per minute.	At 2,000 revolutions per minute.
Hydraulic economy, fraction.....	0.69	0.76
Losses by steam leakage and by the friction of the pillow-blocks, in per centum.....	1.5	1.8
Losses by the friction of the wheels on the steam in per centum	2.5 } 4.0	5.9 } 7.7
Net economy on shaft, fraction...	$0.69 \times 0.96 = 0.662$	$0.76 \times 0.923 = 0.702$
Steam pressure absolute, in lbs. per square inch.....	206.12	
Back pressure absolute, in lbs. per square inch.....	1.16	
Superheating, in Fahr. degrees...	212	
Consumption of steam in pounds weight per hour per theoretical horsepower.....	6.83	
Consumption of steam in pounds weight per hour per effective horsepower.....	$\frac{6.83}{0.662} = 10.32$ lbs.	$\frac{6.83}{0.76} = 9.74$ lbs.

From the above table will be perceived how greatly the economy increases with the speed.

There is given in the above table the calculation of the consumption of steam per hour per effective horsepower, at the two speeds stated.

This consumption, the steam being superheated 212 Fahrenheit degrees, is 10.32 pounds at the speed of 1,500 revolutions of the turbine per minute, which corresponds to about 9.04 pounds of steam per hour per indicated horsepower.

At 2,000 revolutions per minute, the consumption of steam per hour per indicated horsepower will be about 8.60 pounds.

From these figures there appears that the practical results which can be obtained from a multicellular turbine of 1,500 horsepowers, are comparable with those given by the best piston steam-engines that actually exist.

The calculation stated of the economy, it must be well understood, is according to practical coefficients which only practice can give. But in the case of the piston steam-engine, the cylinder condensation is uncertain and often erroneous,

while, in steam-turbines, the continuity of the flow of the steam permits the establishment of a theory of great exactness, applicable to all cases of practice, once the fundamental coefficients are determined.

ECONOMIES OBTAINED WITH DIFFERENT SYSTEMS OF STEAM-TURBINES HAVING MULTIPLE WHEELS.

The analyses that have just been made of the various elements which influence the economy of steam-turbines, enables the calculation to be made of the economies obtainable with different systems of them.

In order to illustrate the preceding considerations by a comparative example, they will now be applied to the two principal systems of reaction turbines having multiple wheels; the turbine with successive falls of pressure, and the turbine with successive falls of speed.

The Rateau multicellar turbine is essentially a turbine with the pressure in successive stages or successive falls of pressure; the speed with which the steam circulates is constant in the different cellules, and as the economy dffers but little from one wheel to another, the work done per wheel remains constant from one extremity of the turbine to the other extremity.

The turbine with the Curtis groups of wheels, is, on the contrary, like the Terry turbine, a turbine with successive falls of speed. The pressure is constant in each group of wheels, but the speed of circulation of the steam, rapidly decreases from one wheel to the other.

Further, if the principle of successive speeds in stages be rigorously applied, such a turbine would act only, as Mr. Mortier has shown, like a single group of wheels for taking up the entire speed of the steam.

As has been previously stated, the Curtis turbine had, at first, 3 wheels in each group, which number has been reduced to 2. The comparison of the Curtis turbine with the multicellular turbine, has been made for the former with two groups of wheels.

In making the comparison of the different turbines, there

is first compared the purely theoretical wheel consisting of a disc furnished with buckets recurved in half-arcs of a circle receiving the steam and permitting it to escape in directions perpendicular to the axis, and at 180 degrees from each other. The turbine with speeds in successive stages, comprises a group of two similar wheels; and the multicellular turbine has only a single wheel.

In both cases, the speed of efflux of the steam, which is to be utilized, will be the same, and, in order to simplify the calculations, will be assumed as equal to unity. The ratio of the peripheral speed of the wheel to the absolute speed of the steam being designated by the letter E, this value will then be that of the peripheral speed. The steam impinges on the wheel with the speed unity; it enters the first bucket with a relative speed W_0 that slackens little by little from its entering to its leaving the canal or passage of the buckets, the coefficient of this reduction being less than unity. The relative speed when leaving is then

$$W_1 = \lambda W_0.$$

From these data, and with the aid of Mr. Rateau's theory previously cited, and which experiment has confirmed, the calculation can easily be made of the total push or pressure on the wheel according to the theorem of the quantities of movement; the work done by the steam is thus immediately deduced, as well as the value of the economy, by the ratio of this work done to the disposable energy in the current of steam at its entrance into the first wheel.

The relative speed of the steam on leaving the first wheel, allows the calculation to be made of its absolute speed at that leaving. This absolute speed becomes reduced during the passage of the steam across the second distributor, and the value of the speed of the steam is thus obtained at its entrance to the second wheel.

The same method enables the push, the work done, and the economy, of the second wheel to be known.

From which can be obtained the aggregate economy of the two wheels.

From these values the maximum economy can easily be calculated by varying the peripheral speed of the wheel, or its ratio to the absolute speed of the steam. The result of the calculation is given in the following table for three values of the coefficient of the reduction of speed of the bucketing: namely, 0.70, 0.75, 0.80.

Absolute speed of the steam $V_0 = 1$.

Ratio of the peripheral speed U of the wheel to the speed of the steam $E = \frac{U}{V_0}$.

	Turbine of action with groups of two wheels.			Multicellular turbine.		
Coefficient of reduction of the relative speed in the bucketing λ	0.70	0.75	0.80	0.70	0.75	0.80
Value of E corresponding to the maximum of work.....	0.233	0.235	0.239	0.50	0.50	0.50
Maximum value of the hydraulic economy.....	0.590	0.642	0.700	0.850	0.875	0.900

The maximum economy that can be furnished by groups of two wheels, is seen to be very inferior to the economy which is given by the multicellular turbine wheel functioning equally in the conditions of the maximum.

This inferiority can be presented in a still more striking manner.

Suppose that the groups of two wheels with successive stages of speed, are transformed into two wheels with successive stages of pressure by separation by means of a diaphragm.

Starting from the figures obtained in practice, the ratio of the peripheral speed of the wheel to the absolute speed of the steam, is 20 per centum. For the same absolute speed of entering the wheel, the fall of pressure being divided in such manner as to give equal work on the two wheels, it is certain that the preceding ratio in the second system rises to more than 28 per centum.

Accordingly, it is easy to calculate that the hydraulic economy of the group with speeds in steps or stages will be 0.688 for the most favorable value of the coefficient $\lambda = 0.80$, while the same group with successively divided pressure will have an economy of 0.767.

In taking for the coefficient of the slowing λ the value 0.70, the hydraulic economy will be 0.580 for the system of successively divided speeds, and 0.704 for the system of successively divided pressures.

Evidently, then, and without taking into the account the faculty which the system with successively divided pressure has of a great deal increasing the number of wheels, which again increases the ratio of the speeds E, and, consequently, the hydraulic economy, the simple introduction of the multicellular principle allows the economy of a turbine with successive stages of speed to be considerably increased.

The preceding reasoning supposes that the bucketing of the turbines is arranged in demi-circumference. In practice this cannot be the case, and, in order to calculate the hydraulic economy it is important to take into the account the angular inclination of the buckets and of the current of steam.

Let there be considered two turbines, one with successively divided speeds, and in groups of two wheels; and the other the multicellular system. The first with a ratio of the peripheral speed to the speed of the steam of 20 per centum, which is the practical case; and the second with a ratio of 35 per centum, which is the case of the Rateau turbine.

Suppose the buckets to have the same inclination in the two turbines, namely, 30 degrees to the plane of the wheel, and that the angle of the steam current or jet is 20 degrees when leaving the distributor.

In taking the total work contained in the steam for unity, there is found that for a value of the coefficient of 0.70, the useful work in the preceding wheel is 0.494, and the useful work in the second wheel is 6 per centum, or the total of 0.554 for the two wheels. In the Rateau turbine, on the contrary, the useful work in the first wheel is 0.700, and that due to the remaining speed of 6 per centum, making a total of 0.760.

In these conditions, the Rateau turbine gives a supplement of 37 per centum of work.

If, on the contrary, the supposition be made that the coefficient λ is more favorable and attains 0.80, a value never yet obtained in practice, there will be found that the work utilized in the two wheels with speed successively divided is 0.646, and in the wheels with pressure successively divided 0.783, or a supplement of still 21 per centum.

This last discussion, like the preceding one, leaves no doubt on the primordial fact that the hydraulic economy of a turbine with successive falls of pressure, multicellular, is always superior to that of a turbine with successive decreases of speed in the steam.

COMPARATIVE ECONOMY IN FUNCTION OF LOAD.

In Figure 15 are reunited the comparative curves of the total economy in function of the load, and at constant speed, of a certain number of groups of generators of electricity by means of different systems of steam-turbines.

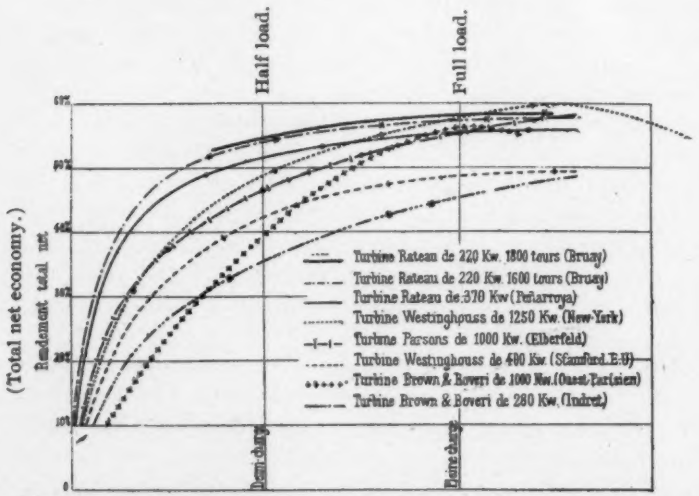

Half load.
Full load.
(Total net economy.)
Rendement total net.
Turbine Rateau de 220 Kw. 1800 tours (Bruay)
Turbine Rateau de 220 Kw. 1600 tours (Bruay)
Turbine Rateau de 370 Kw (Peñarroya)
Turbine Westinghouse de 1250 Kw. (New-York)
Turbine Parsons de 1000 Kw. (Elberfeld)
Turbine Westinghouse de 400 Kw. (Stamford. E.-U)
Turbine Brown & Boveri de 1000 Kw. (Oued-Zarsien)
Turbine Brown & Boveri de 280 Kw. (Indret)
Demi-charge
Pleine charge

Fig. 15—COMPARISON OF THE TOTAL ECONOMY OF TURBINE-DRIVEN ELECTRICAL GROUPS, IN FUNCTIONS OF THE LOAD.

These curves have been traced by calculating the economy according to the experimental results that have been published. The abacus has been used for that purpose.

In a general way, an examination of these curves makes known that the total economy increases in function of the load until a certain maximum is attained; beyond which point it decreases.

They include only two groups of Rateau turbines: one of 220 kilowatts at low pressure, and the other of 400 kilowatts at normal pressure.

The other curves belong to different turbines of the Parsons system, constructed either by the inventor or by the firms Brown-Boverie, and Westinghouse.

The economy of the Rateau groups is generally superior, for equal power, to that of the other machines, and with feeble loads, the curve is maintained in the upper regions.

It is scarcely necessary to say, that these curves do not exactly give the real economy of the turbine separate from its generator of electricity, because they include, as is well understood, the economy of that generator.

Nevertheless, from the practical point of view, the important fact is the net economy of the turbine and of the dynamo or alternator when combined.

An other remark that results from these curves is that the steam-turbines have generally a satisfactory economy with feeble loads.

When the steam is throttled and its pressure consequently lowered, the hydraulic economy of some of the wheels of the turbine increases, and therefore recuperates a more important proportion of the work, although the disposable energy in the steam was diminished.

On the other hand, the frictions of the wheels on the steam diminish with diminution of the mean density of the steam.

These phenomena do not happen with the piston steam-engines, and, consequently, their economies with feeble loads are inferior to those of steam-turbines.

Figure 16 gives comparative curves of the maximum econ-

omy at full load for different systems of turbines. The abscissas represent the value of the full load up to 4,000 kilowatts; the ordinates represent the total economy at full load.

The appearance of these curves is very different for the different systems. There must not, however, be drawn too precise consequences from the calculations made by means of

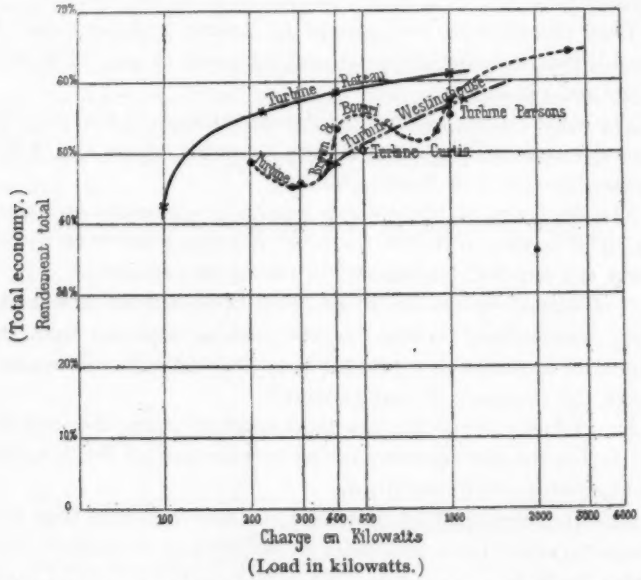


Fig. 16.—MAXIMUM ECONOMY OF ELECTRICAL GROUPS WITH DIFFERENT SYSTEMS OF TURBINES.

the too scanty information published for the different machines, and which should be completed whenever the data furnished by their constructors shall be more numerous.

APPLICATIONS OF THE RATEAU TURBINE.

I.—GROUPS FOR THE GENERATION OF ELECTRICITY.

The most important application made of the Rateau turbines up to the present time, has been for the operation of generatrices of electricity either with continuous current or with triphased current.

CONTINUOUS CURRENT.

Dynamos of continuous current operated directly by steam-turbines, have quite special difficulties of construction. Their great angular velocity compels their establishment with a small number of poles; whence results that the reaction of the introduction is considerable, and that the employment of special systems of compensation is necessary to prevent the disastrous effects of a bad commutation of the brushes.

With the Parsons turbine, the firm of Brown has principally employed a system of compensation formed by supplementary spirals placed in the polar pieces and fed or supplied in series.

The workshops of Sautter, Harlé and Co. have had recourse to a system of complementary inducting poles, fed in series, and creating a magnetic field in the plane in which the commutation is made.

This first difficulty being overcome, the dynamos offer another, which is the construction of the collectors. This delicate part of the machine should be made with the minutest care, because it is composed of heterogeneous pieces of metal and of isolants which are very difficult to center and to adjust rigorously for preventing the effects due to the centrifugal force.

The centrifugal force is, in fact, considerable in generatrices operated by steam-turbines; it attains, and often surpasses, at the periphery, 2,000 times the weight.

Also, the equilibrating of all the revolving parts should be minutely cared for. Special arrangements for the statical and for the dynamical equilibrating have been foreseen.

With dynamos of continuous current, the vibrations can render the functionment quite impossible by reason of the sparks they produce in the brushes if the latter momentarily quit the collector.

The peripheral speed of the collectors being considerable, the effect of the sparks is a great deal more injurious than in the ordinary dynamos, and it becomes indispensable to nearly completely avoid fire at the brushes.

Generally, the brushes are metallic, but in certain special

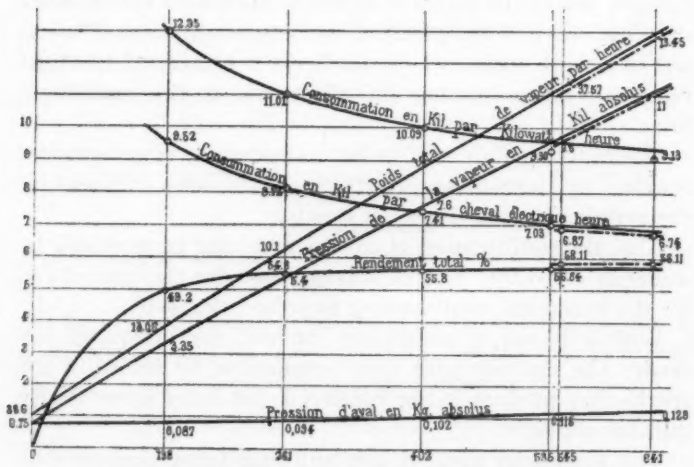
cases, notably with turbines fed by low pressure turbines where the angular velocity is much less, it is possible to employ brushes of carbon.

Always, as well with brushes of carbon as with metallic brushes, perfect equalibration of the moving parts is indispensable.

The designs of the construction of the new generators of electricity operated by Rateau turbines, have been made under the direction of Mr. Sacquet, Engineer of the Electrical Service in the establishment of Sautter, Harlé and Co.

Figure 18, Plate V, represents a group of 500 electrical horsepowers with continuous current: the turbine made in two bodies operates directly two dynamos placed end to end and arranged for three wire distribution.

Several of these machines have been made for the metallurgical works of the Society of Pennaroys, in Spain. Their rotary speed is 2,000 revolutions per minute.



(Power in electrical horsepowers measured at terminals.)

Fig. 17.—TURBINE DYNAMOS OF 500 HORSEPOWERS. CURVES OF PRESSURE, CONSUMPTION AND ECONOMY, AT A CONSTANT SPEED OF 2,400 REVOLUTIONS, IN FUNCTIONS OF THE POWER.*

* The curves ——— . ——— . ——— correspond to the operation with steam superheated about 10° cent.

The curves of consumption of steam, and of economy, are shown in Figure 17 which gives the results of the trials. It will be observed that at the speed of 2,400 revolutions per minute, the group was able to furnish up to 640 electrical horsepowers which represent about 700 effective horsepowers on the shaft. The total economy rises to 58 per centum with the steam slightly superheated.

The curves also show the remarkable proportionality of the total weight of steam consumed to the useful load; this weight of steam being sensibly represented by a straight line in function of the load.

With vacuum, when the machine does not do any external work, the dynamos being excited, the consumption of steam represents 9 per centum of its consumption at full load, a remarkably low figure showing how feeble are the frictions in the Rateau turbine.

An analagous group of 500 electrical horsepowers, operated by a piston steam-engine, consumed with vacuum 20 to 22 per centum of the weight of steam consumed at full load.

The consumption trials which have enabled the curves in question to be drawn, were made by Mr. Wissler, Engineer of the technical investigation department of the previously mentioned workshops, under the direction of which the plans were made for the execution of the different Rateau turbines manufactured in them.

There can also be remarked concerning these curves, the proportionality of the absolute pressure of the steam when entering the turbine to the useful load.

The turbine can be said to be a true counter of the consumption of steam, and in order to know the consumption at any given load, all that is required is to have measured the consumption for two other loads, the curve being sensibly a straight line.

The following table shows the comparative results of calcutions made by Mr. Rateau for a group of 500 horsepowers, and the results of trials with full load. It shows, also, what would be the consumption by the same turbine per electrical horse-

power, functioning with a higher pressure of steam and a better vacuum.

It also shows for this group the increased economy for higher powers.

A RATEAU TURBO-DYNAMO OF 500 ELECTRICAL HORSEPOWERS.

——	Results of the calculations made by Mr. Rateau.	Results of the experiments made.	Results under the most favorable conditions.
Absolute pressure of the steam, in pounds per square inch	142.223	136.541	213.345
Absolute pressure of the exhaust, in pounds per square inch	2.844	1.636	1.159
Revolutions per minute	2,000.000	2,000.000	2,400.000
Consumption of steam, in pounds weight per hour per electrical horsepower	17.637	15.653	11.861 (Per kilowatt, 16.138)
Electrical horsepowers developed	500.000	510.000	530.000
Economy, total net	0.550	0.563	0.581

A general section of the 500 horsepowers turbine in two bodies, is represented in Figure 18. It shows the entrance of the steam, and the exhaust, for the two bodies; also, the arrangement of the pillow-blocks, all of them being outside

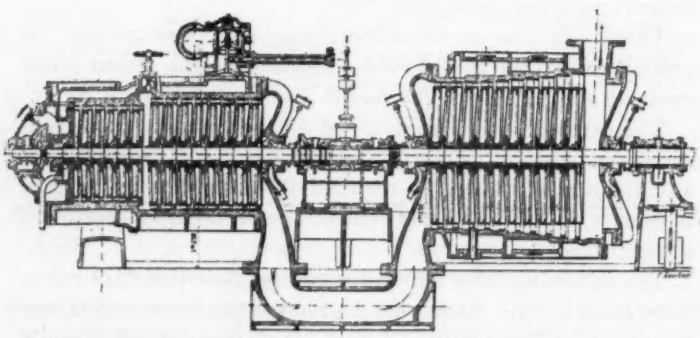


Fig. 18.—Section Through a Rateau Turbine of 500 Horsepowers.

of the casings or shells. The lubrication is made in the ordinary way, the forcing of the oil upon the journals, as originally done by special pumps, not being found necessary.

The division of the turbine into two bodies acting upon the same shaft is not necessary: on the contrary, the two bodies can be placed on two parallel shafts, one on each shaft, the two bodies being connected by an intermediate pipe conveying the steam from one to the other.

This arrangement is represented in Figure 19, Plate V, for an ensemble of two connected 500 horsepowers bodies upon two parallel shafts functioning in a steel works in Russian Poland.

The two dynamos are combined in quantity, while the two bodies are in series.

The functionment of this type of turbine is as satisfactory as that of the equivalent turbine in a single body. The first cost is a little greater, but it has the practical advantage of allowing the separate functionment of one of the bodies with its dynamo, in the case of an accident to the other.

In regard to space occupied, according to the case, there is an advantage in employing the group with a single shaft, or the group with two shafts.

The utilization of two dynamos permits, especially as regards the distribution of the force in a workshop, the employment of the three wire system which enables a sensible economy to be realized in the cost of the canalizations.

The working of dynamos by turbines is particularly well adapted to the distribution with three wires.

TRIPHASED CURRENT.

The construction of generatrices with triphased current in relation to turbine, presents less difficulty than dynamos of continuous current. The suppression of the collector is an important advantage as it permits in practice the possibility of higher angular speeds for the alternators than for the dynamos of continuous current.

Sautter, Harlé and Co. have tried successively three types of

alternators: the turning iron alternator; the turning inducing alternator; and the turning inductor alternator.

The turning iron alternator would be from the mechanical point of view, the ideal generatrice for the steam-turbine, did not its very construction restrict the conductor to only two polar portuberances equivalent to four alternate poles. Now when it operates a machine giving 50 periods per second, evidently the speed of rotation corresponds to 1,500 revolutions per minute; but, for turbines of less than 1,000 horsepowers, the speed of 1,500 revolutions per minute is too small and leads to a turbine of disproportionate dimensions and of mediocre economy.

But this is not the only disadvantage of the iron turning system: with equal weight, the magnetic leakage and the fall of tension in these machines are more considerable than for the other types of alternators.

With equal quality, the weight of these machines is notably greater.

These disadvantages are very regretable, because the turning iron alternator, allows the construction of an entirely massive moving part that can easily be perfectly equilibrated at the moment of its construction, and which has no chance of being injured.

The second type of alternator, is the turning inducing alternator. It is according to this type that Parsons has constructed the greatest number of electrogen groups functioning in England.

The turbo-alternator of the town of Elberfeld, well known by the complete experiments made with it, is properly of the same type.

With a turning inducer, a machine of minimum weight can be constructed of which the inducing reaction is the feeblest, being given as the inductors can be saturated.

But there is a difficulty; that of constructing the inducers, which have to turn for high tensions at very considerable peripheral speeds; the place lacking for the isolants, and the

effects of accumulation due to the centrifugal force, can, after a while, compromise the isolation.

The turning inductor alternator comprises an inducer fixed in sheet iron, and a polar turning inductor, constructed in such a manner as to be able to resist great centrifugal forces.

The tension of the current in the inductor being feeble, since it acts only as the current of excitation, a good isolation can be more easily obtained.

The disadvantage of this system consists in the small number of its poles. For a frequence of 50 periods, if it be desired to make 3,000 revolutions per minute, only two poles are necessary; at 1,500 revolutions per minute, four poles are sufficient.

These numbers of poles, for a frequence of 40 periods, correspond to 2,400 and to 1,200 revolutions per minute.

Now the alternators having a small number of poles, have a strong inducing reaction, notably for the bipolar alternators, the induction cannot be saturated, because the magnetic losses would become too large.

The fall of tension in the alternators of two or four poles is then considerable when passing from working at a vacuum to working with a full load and constant excitation. It becomes necessary, when the variations of the load are frequent on the iron net-work that feeds the machine, to provide an automatic system of compensation of the voltage, and above all for the bipolar alternators.

Figure 19 represents a group of 1,000 electrical horsepowers, 700 kilowatts at 1,000 volts and 1,500 revolutions per minute. The alternator is of iron and turning, and the turbine is of the plan first designed, consequently with interior pillow-blocks.

This machine which was designed and constructed in 1900, has given excellent results from the mechanical point of view, but the defects of the iron turning alternators are very sensible: great fall of tension; great magnetic leakage.

The turbine is of the same design as the one made for the Torpedo Boat *243*, of which more will be said later on.

The curves given in Figure 20 show the economy of that

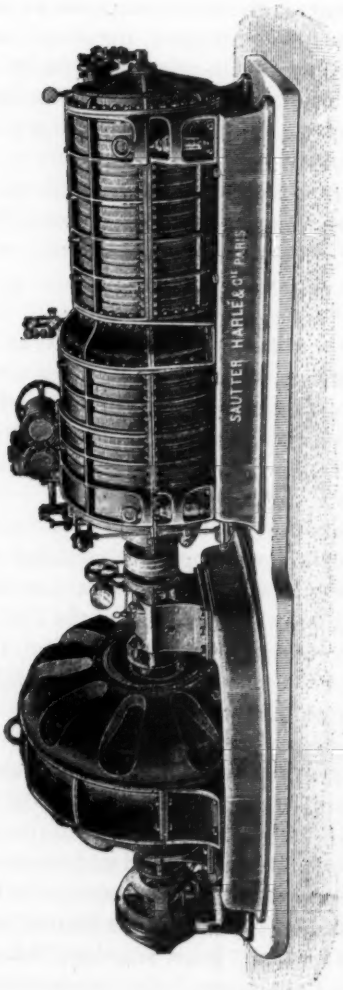


Fig. 19.—Turbo-Alternator of 1,000 Horsepowers.

design of turbine at the speed of 1,500 revolutions per minute. This economy has been measured according to the consumption of steam and according to the measure of the economy of the alternator itself, calculated according to a method devised by Engineer Blondel.

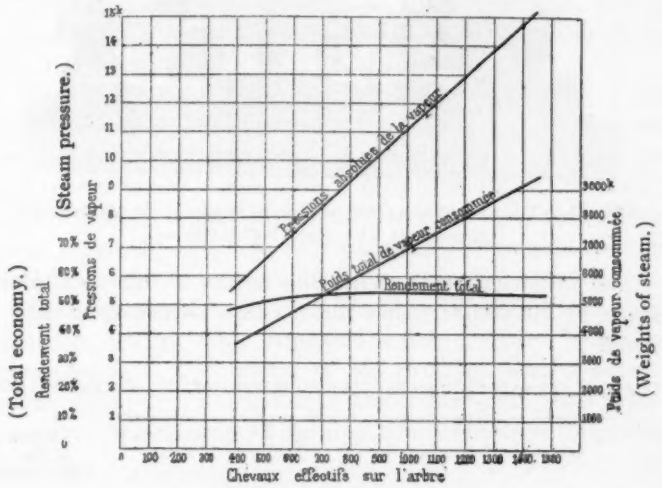


Fig. 20.—RATEAU TURBINE OF 1,000 HORSEPOWERS. TOTAL ECONOMY AND STEAM CONSUMPTION.

The design of the turbine in question, was made for a speed of 1,800 revolutions per minute: the curves of economy relate to the speed of 1,700 revolutions per minute.

This type was too much reduced as regards dimensions, because of the application that was desired to be made of it in marine propulsion; but, nevertheless, it is interesting to find that the result obtained with full load during the experiments in 1903, correspond to a value of 0.535 as the net economy of the turbine, while the calculations made by Mr. Rateau in 1898 assigned an economy of the value of 0.530.

Figure 21 represents a group of 400 electrical horsepowers with 5,500 volts at the speed of 3,000 revolutions per minute.

This machine is now functioning in a station of "The Electrical Company of the Loire."

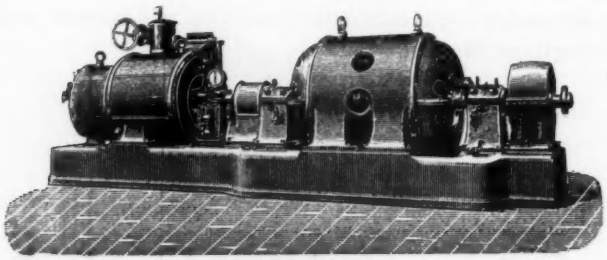


Fig. 21.—TURBO-ALTERNATOR OF 400 ELECTRICAL HORSEPOWERS, 5,500 VOLTS AND 3,000 REVOLUTIONS.

The following table shows the results of the calculations made by the builders, and the results obtained from experiment.

TURBO-ALTERNATOR OF 400 ELECTRICAL HORSEPOWERS.—RATEAU SYSTEM.

	Calculated by the builders.	Results of the experiments.
Pressure of the steam, in pounds per square inch above zero	170.678	168.403
Pressure of the exhaust, in pounds per square inch above zero	2.133	2.845
Revolutions per minute	3,000.000	3,000.000
Electrical horsepowers developed	370.000	388.000
Economy	0.470	0.487
Consumption of steam, in pounds weight per hour per electrical horsepower	18.739	19.224
Consumption of steam, in pounds weight per hour per kilowatt	25.573	26.014

The economy obtained would have been increased by augmenting a little the dimensions of the turbine which contains only a dozen working wheels; but is still superior to that of any group of turbines of the same power of any other system.

It is interesting to observe that this machine debits upon a wire net work whose factor of power is very feeble, not surpassing as a mean from 0.55 to 0.60. This net work feeds

small synchronous motors employed in the ribbon factories at Saint-Étienne.

The alternator being bipolar, and the factor of power being very feeble, the fall of tension with constant excitation between the vacuum and the full load is considerable; in fact, it exceeds 40 per centum. Under these conditions, it is indispensable to compensate this fall by a compounding of the alternator.

This compounding has been realized by means of a dispositive designed by Engineer Blondel which consists of the employment of an excitatrice of reaction and double excitation combined in series with a transformator. The results thus obtained are very satisfactory, and the compounding has been shown to be made with a variation of only some hundredths on the measured voltage.

At the present date Sautter, Harlé and Co. have under construction groups with triphased current of 300 electrical horsepowers at 1,000 volts; of 360 electrical horsepowers at 2,000 volts; of 700 electrical horsepowers at 5,000 volts; constructed with induced turning alternators or with turning inductor.

CONSTRUCTION OF RATEAU TURBINES IN THE WORKSHOPS OF OERLIKON.

The construction of electrogen groups, with turbines of the Rateau system, has likewise been undertaken in Switzerland by the workshops of Oerlikon.

Several groups with generators having a continuous current or a triphased current have been constructed in these well known shops.

A great deal of talk has been made of certain difficulties encountered at the commencement by the Oerlikon shops during the first constructions.

These difficulties arise, on the one part, during the construction of new electrical machines which require, as has previously been stated, the solution of problems not found in the construction of slower machines.

On the other part, particular circumstances, such as the arrangement for the entrance of the steam into the turbine, and those of the piping, have been in the Rheinfelder workshop, the cause of incidents that have been much exaggerated by persons of opposing interest.

At the present time, the Oerlikon workshops have overcome these first difficulties, and have perfected the manufacture of steam-turbines of the Rateau system. There can be cited, notably, an electrogen with continuous current, having a power of 100 kilowatts with 3,000 revolutions per minute, installed at Ruti in Switzerland, and which has been functioning normally during 18 months.

It is important to recall that the Rateau turbine with two bodies in cascade (Figure 16, Plate IV), having 2,000 horsepowers, which has given excellent results during recent experiments, in the propulsion of a torpedo boat constructed by Yarrow in England, came from the Oerlikon workshops.

Later on, remarks will be made concerning the application of the turbine to marine uses.

II.—TURBO-PUMPS.

PRINCIPLE OF THESE APPARATUS.

The working of a centrifugal pump by a steam-turbine, seems, at the first view, to present insurmountable obstacles.

As has been shown by Mr. Rateau, in his memoir published in the "Bulletin de la Société de l'Industrie Minerale" in 1902, it is always possible to operate a centrifugal pump at considerable angular velocity by a steam-turbine, but, unless certain precautions are taken, the economy of such a combination, will be very inferior.

Beyond a certain angular speed, the blade of a centrifugal pump is not capable of delivering, the receiving itself being prevented.

This phenomenon, which is comparable to that of cavitation in the case of the blades of a propelling screw, is caused by the discontinuity in the stream of liquid which can no longer follow up the moving blades of the centrifugal pump. Beyond

a certain debit for a given speed and a given delivery, the economy of the pump continues to decrease and finally becomes altogether defective.

The phenomenon of cavitation obliges the loading of the wheel of the centrifugal pump whenever its angular speed surpasses a certain limit for a given debit and a given height of elevation of liquid.

When the intention is to directly work the centrifugal pump by a steam-turbine, it is indispensable to load this pump by means of an auxiliary pump which produces the normal debit with a great deal less height of elevation, attaining at the most 5 to 10 per centum of the total height to be produced.

When turbines of a single wheel are employed like those of Laval, or like those which Rateau first constructed, it becomes impossible, owing to their great angular speed, to load the principal pump by placing the auxiliary pump upon the same shaft. The phenomenon of cavitation, suppressed for the principal pump is reproduced for the auxiliary pump.

For this reason, the first inventors discovered that it was necessary to reduce the angular speed of the auxiliary pump, and consequently to place on the shaft of the principal pump reducing gearing.

In this manner Laval's pumps, and the pumps of other analogous machines, are arranged.

Figure 20, Plate V, represents the first Turbo-pump invented by Mr. Rateau, and with which experiments were made in 1901.

The turbine, formed of a single Pelton wheel, had the speed of from 15,000 to 18,000 revolutions per minute, and operated a centrifugal pump of only 3.15 inches diameter.

This wheel was fed, under load, by means of another pump; the height to which the water was lifted was 997½ feet with a debit of 882.89 cubic feet per hour, and an economy, pump and turbine combined, of 0.280. The wheel would absorb under these conditions about 50 horsepowers at its axis.

In another experiment, there was debited as high as 1,518.57

cubic feet per hour, at 862.87 feet of height, and the pump then absorbed 70 horsepowers at its shaft.

These remarkable experiments demonstrated at once, that a centrifugal pump could deliver, so to speak, to illimitable heights, and that the economy of such a combination was comparable to that of the similar groups of the best pumping piston steam-engines.

By total economy of a turbo-pump, is meant the relation of the useful energy in the water lifted, to the total energy in the steam for the fall utilized from the highest pressure in the turbine to the pressure in the condenser.

The experiments that have just been cited show the possibility of making turbo-pumps of excellent economy.

Mr. Rateau has conceived the idea of operating by means of multicellular steam-turbines, multicellular pumps which he has designed.

The Rateau pump is essentially composed of one or two bodies containing a series of movable wheels and of fixed diffusers, allowing the successive increase of the pressures for each wheel in order to reach the total pressure.

The employment of such a combination permits, for the same angular speed, the suppression of the phenomenon of cavitation, or, at least, a restriction of its limits.

On the other hand, by working the pump, thus combined, by a multicellular turbine, the speed, for the same conditions of employment, can be notably lowered, and the auxiliary pump can be placed on the same shaft as the principal pump.

DESCRIPTIONS OF DIFFERENT TURBO-PUMPS.

Figure 22 represents a turbo-pump delivering 6,356.81 cubic feet of water at a height of 688.99 feet, and absorbing at its shaft 230 horsepowers.

The turbine exhausts into an ejecto-condenser with a vertical axis; this condenser is in the same casing with the turbine. The condensing water is supplied by a small centrifugal pump by the main shaft itself.

The ensemble of this apparatus constitutes thus a small

complete system for the elevation of water, with steam-motor, condenser, the lifting pump, and the auxiliary pump.

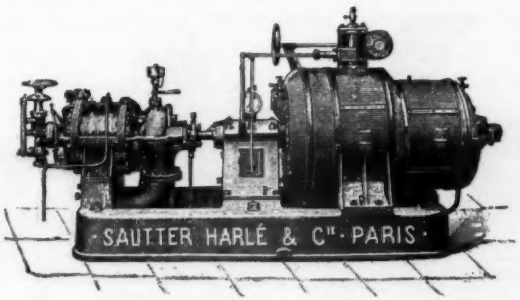


Fig. 22.—RATEAU TURBO-PUMP OF 230 HORSEPOWERS.

The following table gives the comparative results of the calculations made by the builders, and the results given by experiment.

The total net economy rises to 0.366, corresponding to a consumption of steam of 28.109 pounds per hour per useful or net horsepower at the low pressure of 90.316 pounds per square inch absolute.

RATEAU TURBO-PUMP RAISING PER HOUR 6,356.81 CUBIC FEET OF WATER 688.99 FEET.

	Calculated by the builders.	Results of the experiments.
Cubic feet of water delivered per hour	6,356.810	6,356.810
Total height of the delivery, in feet	675.865	682.427
Revolutions per minute	3,100,000	3,250.000
Pressure of the steam, in pounds per square inch above zero	128.007	104.539
Pressure of the steam in the condenser, in pounds per square inch above zero	2.560	2.475
Total net economy	0.344	0.366
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water (condensation included)	28.219	28.109

A more powerful Rateau turbo-pump has been recently constructed for the Société des Mines de Bruay, in the Pas-de-Calais.

This machine, represented in Figure 23, can raise per hour 8,828.900 cubic feet of water 1,181.124 feet high.

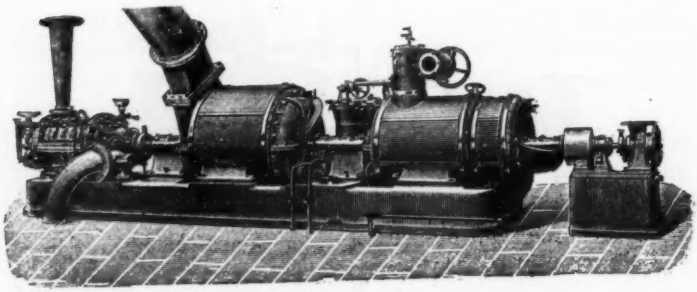


Fig. 23.—RATEAU TURBO-PUMP WITH CAPACITY OF 8,829 CUBIC FEET OF WATER AGAINST A HEAD OF 1,181 FEET.

The precisions of the calculations, and the results of the experiments are shown in the below table.

RATEAU TURBO-PUMP RAISING PER HOUR 8,828.900 CUBIC FEET OF WATER 1,181.124 FEET.

——	Calculated by the builders.	Results of the experiments.
Cubic feet of water delivered per hour..	8,828.900	9,393.950
Total height of the delivery, in feet.....	1,181.124	1,194.248
Revolutions per minute.....	2,250.000	2,200.000
Pressure of the steam, in pounds per square inch above zero.....	99.561	100.983
Pressure of the steam in the condenser, in pounds per square inch above zero.....	1.561	1.621
Total net economy.....	0.430	0.425
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water (condensation not included).....	22.046	22.707

The great economy of this pump is remarkable, as, everything included, it rises to 0.425. Further on will be seen, in relation to the consumption of steam, a total economy of over 40 per centum when steam of high pressure is used.

One of the great advantages of turbo-pumps, in addition to their excellent economy, consists in their small bulk comparably with the bulk of piston-pumps.

There are shown in Figure 24, the comparative dimensions of the pit necessary for the turbo-pump in question, and necessary for the pit of a piston-pump of the same power.

The turbo-pump of the Bruay mines represents, in fact, 500 horsepowers on the shaft, but the dimensions of the turbine with two bodies and those of the pump itself, are so greatly below the dimensions of a corresponding piston-pump that there results an economy in the dimensions of the pits and in the volumes of the masonry nearly in the proportion of 1 to 10.

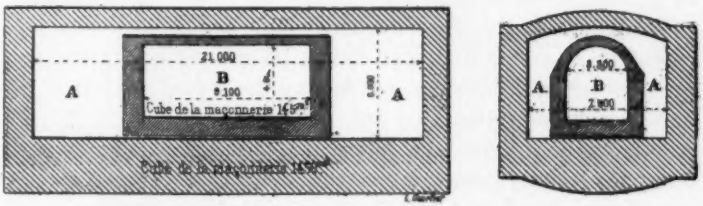


Fig. 24.—A, Steam Piston Pump. B, Rateau Turbo-Pump.

Another turbo-pump functions in a metal-mine, and delivers per hour 3,884.716 cubic feet of water at the height of 229.663 feet.

This turbine is constructed to receive the steam at the pressure of 57 pounds per square inch above the atmosphere, but it really functions at the pressure of only 14.223 pounds per square inch above the atmosphere, the delivery being notably less than that for which the apparatus was calculated.

With the turbo-pump very complex conditions can also be realized.

Figure 21, Plate VI, represents an apparatus constructed for a coal mine in Russia. It first raises 3,178.400 cubic feet of water per hour 295.281 feet high during the first period, then, during the second period, it raises this water 524.944 feet high.

The problem was solved by constructing a pump with three wheels, of which only two were used to elevate the water to

the 295.281 feet height; the farther height of the 524.944 feet was effected by adding the third wheel and slightly increasing the speed of the turbine.

TURBO-PUMPS FOR FEEDING.

The turbo-pumps of the Rateau system can be advantageously applied to the feeding of boilers.

When the service is the feeding of a group of large boilers, it can be done by a turbo-pump with greater convenience and greater economy than by the auxiliary steam-pumps worked by a piston-cylinder as usually employed.

While, ordinarily, these auxiliary feed-pumps with piston-cylinders, consume from 176 to 264 pounds weight of steam per hour per useful horsepower forcing the feed-water into the boilers, a consumption that can rise to 330 pounds weight of steam per hour when the pump delivery is low, the turbo-pump for feeding does not consume more than from 44 to 55 pounds weight of steam per hour per useful horsepower.

This small consumption of steam has a greater importance than appears at first view; for at the moment the boiler is required to furnish a large quantity of steam in a given time, the quantity of steam withdrawn from it for feeding is the largest, and is then the most needed for other use.

Also, notwithstanding that its money cost is a little greater, the turbo-pump is notably the cheapest, if account be taken of the cost of the steam consumed.

Among the numerous types of steam-pumps for feeding, there can be cited a feeding turbo-pump delivering 1,412.6 cubic feet of water per hour at the height of 393.7 feet; that is to say, under the pressure of 170.7 pounds per square inch above the atmosphere, installed in the mines of Lens.

A system of automatic regulation permits this apparatus to function continuously even when its delivery is reduced to zero.

Its management by the fireman, consists in gradually opening, or gradually closing, the feed-valve.

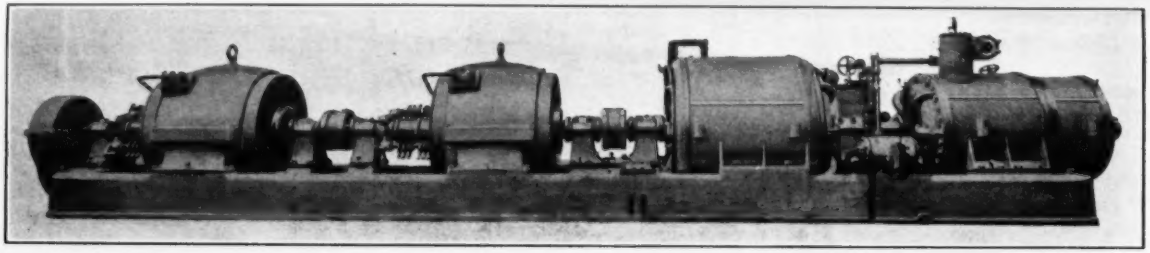


Fig. 18.—Group of 500 Horsepowers, Continuous Current.

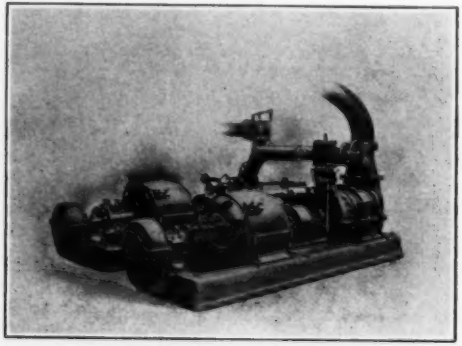


Fig. 19.—500-Horsepower Group.

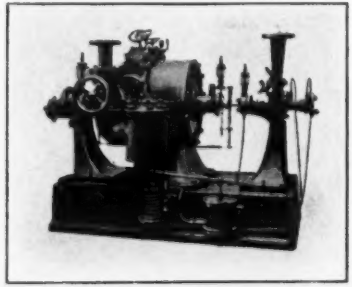


Fig. 20.—Turbo-Pump, Rateau (1901).

PLATE V.

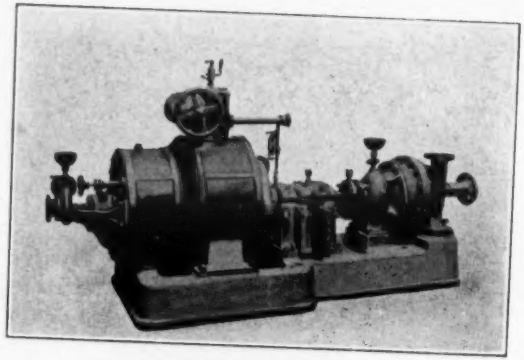


Fig. 21.—Turbo-Pump.

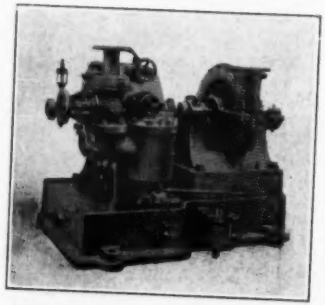


Fig. 22.—Turbo-Ventilator, Rateau (1901).

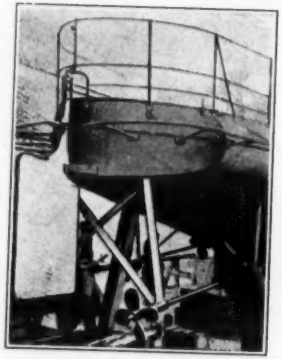


Fig. 23.—Stern of Torpedo Boat 243.

Fig. 24.—Torpedo Boat 243 Under Way.

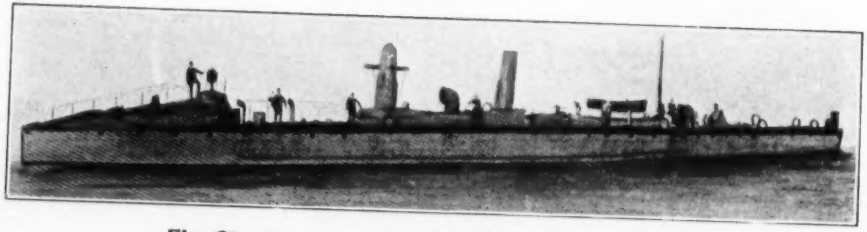


Fig. 25.—General View of Yarrow Torpedo Boat.

PLATE VI.

TURBO-PUMP WITH LARGE DELIVERY, FOR SUPPLYING POTABLE WATER TO CITIES.

The Rateau turbo-pumps, both on account of their economy, and on account of the mechanical excellence of their functionment, are especially adapted for supplying cities with water under great pressures.

The following figures show the results that have been obtained with one of these apparatus under conditions analogous to those found in the largest cities of Europe.

Delivery of water, in cubic feet per hour	49,441.84
Total height of the delivery, in feet	459.33
Useful horsepowers applied to lifting the water	720.00
Total economy, condensation included	0.46
Pressure of the steam, in pounds per square inch above zero	213.34
Pressure of the steam in the condenser, in pounds per square inch above zero	1.14
Superheating of the steam, in Fahrenheit degrees	212.00
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water	15.09

If the above consumption be referred to the indicated horsepower, and the ratio of the useful horsepower to the indicated horsepower be taken as 88 to 100, the consumption of steam in pounds weight per hour per indicated horsepower will be 13.28.

Hence, the system in question is not inferior in any respect to the best piston-pumps actually in use.

III.—TURBO-VENTILATORS.

The great angular speeds of steam turbines render them particularly adapted for the direct working of ventilators, compressors, or blowers against high pressures.

There will be found in Mr. Rateau's memoir* already cited, the experiments made on a turbo-ventilator of his first design, represented in Figure 22, Plate VI.

This apparatus consisted of a one wheel steam-turbine of the Pelton system working directly a ventilator wheel of 12 inches diameter.

* Ventilators and centrifugal pumps for high pressures. Bulletin de la Société de l'Industrie Minérale, première livraison, 1902.

PLATE VI.

Fig. 24.—Torpedo Boat 243 Under Way.

At the speed of 20,200 revolutions per minute, this ventilator delivered 23.167 cubic feet per second under a pressure of a column of water 18.7 feet high. The useful work done in compressing the air was about 40.44 horsepowers, and the total economy of the group was 0.307, the pressure of the steam when entering the turbine being 147.9 pounds per square inch absolute.

Since that epoch, the system in question has been perfected, and a number of this apparatus has been constructed for various applications.

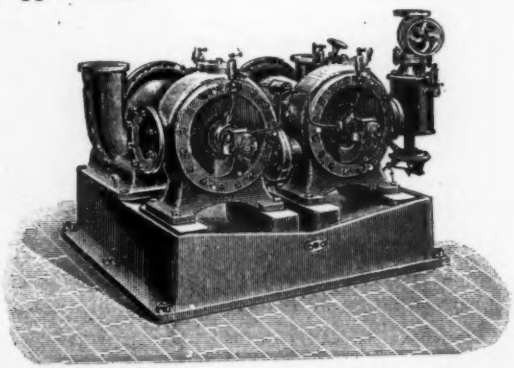


Fig. 25.—RATEAU TURBO-VENTILATOR FOR BLAST FURNACES.

One of them is employed in a sugar refinery for compressing the carbonic acid extracted from burnt lime; it is also employed in the arsenals for various purposes.

That apparatus exerts about 60 horsepowers on the shaft. An ingenious system of regulation acting upon the turbine, enables the pressure of the compression to be constantly maintained, be the delivery what it may.

The design shown in Figure 25 has been executed for the blast of a cupola in the "Forges de Chatillon et Commentry": it delivers 4,944 cubic feet per minute under the pressure of 10 inches of mercury. The compression of the air requires about 100 horsepowers.

The following table gives the results of the calculations

made by the builders, and also the results of two series of different experiments.

RATEAU TURBO-VENTILATOR DELIVERING 70.631 CUBIC FEET OF AIR PER SECOND UNDER THE PRESSURE OF A WATER COLUMN 6.562 FEET HIGH.

	Calculated by the builders.	Results of the experiments.	
		1	2
Delivery of air per second, in cubic feet	75.929	78.401	56.858
Maximum pressure, in feet high of water column	7.546	8.858	11.713
Revolutions per minute	13,800.000	14,600.000	14,600.000
Pressure of the steam, in pounds per square inch, absolute	85.338	99.561	106.672
Pressure in the condenser, in pounds per square inch, absolute	2.845	2.845	4.381
Total net economy	0.350	0.362	0.377
Consumption of steam per hour per useful horsepower compressing air, in pounds weight	20.296	20.851	19.870

The total economy of the above apparatus rises to the remarkable figure of 0.377.

This steam-turbine is composed of two wheels of the Pelton system joined in series, each working a distinct shaft. On each shaft a ventilating wheel is placed, giving the total pressure and one half of the delivery. The orifices of the delivery are joined in quantity.

The principle of the apparatus permits the construction of machines of still greater power, replacing advantageously the usual blowing engines.

To be convinced of this latter, consider Figure 26 representing the spaces occupied by the apparatus in question and by the equivalent usual blowing engine.

Another design, now in construction, gives a delivery of 3,445 cubic feet of air per minute under the pressure of a column of water 13 feet high.

Finally, in a few months will be finished the construction of a turbo-compressor of 350 horsepowers at the shaft, compressing the air to the pressure of 85 pounds per square inch

above the atmosphere. This apparatus is worked by a low pressure turbine utilizing the exhaust steam of a piston steam-engine, pumping water from a mine.

The turbine is in two high pressure bodies allowing it to function alone during the hours that the pumping engine is not working.

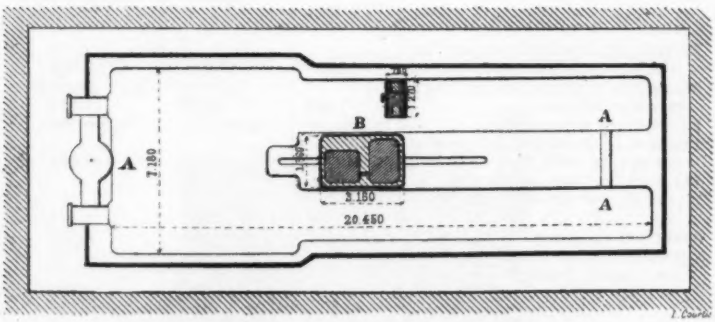


Fig. 26.—A, Ordinary Blowing Engine (1,765.3 Square Feet). B, Rateau Turbo-Ventilator (75.3 Square Feet).

This, quite a special branch of manufacture of the application of steam-turbines to the compression of air, has a great future, not only as regards mines but also for metallurgical operations, the Rateau apparatus advantageously replacing all other blowing engines of all kinds in steel works and in smelting.

LOW PRESSURE STEAM-TURBINES.

The utilization of the exhaust steam escaping from steam-engines into the atmosphere, has been the object of many designs made by Mr. Rateau.

The excellent economy of turbines using low pressure steam has enabled him to show that the utilization of this exhaust steam, impossible with piston steam-engines, is easily accomplished by means of additional steam turbines.

If steam be taken of atmospheric pressure, representing about 26.6 cubic feet per pound weight, and if this steam be expanded to the pressure in the condenser, or to about the

pressure of 1.4 pounds per square inch absolute (see the abacus), there will be found that the weight of steam consumed per theoretical horsepower is about 17 pounds per hour.

Now, the multicellular steam-turbine in proper adjustment and utilizing the steam in these conditions, has an economy surpassing 60 per centum, and can attain even 65 per centum.

The effective horsepower upon the shaft under these conditions, represents a consumption of about 26.8 pounds weight of steam per hour.

If the turbine be replaced by a piston steam-engine, not only will the dimensions of this motor be, for equal power developed, enormously disproportionate, but its economy, by reason of its frictions and thermic losses, will be extremely low.

The dimensions of the turbine, on the contrary, are far from increasing in proportion to the volume of the steam to be utilized. Only the sections of the distributors should be calculated so as to allow the total flow of steam to pass at a convenient speed.

The steam turbine, therefore, as opposed to the piston steam-engine, is in comparatively the same conditions as the centrifugal pump is to the piston-pump.

Struck by these advantages, Mr. Rateau sought to combine the turbines with stationary steam-engines without condensation; that is to say with piston steam-engines pumping out mines, and piston steam-engines working rolling mills.

The powers which the apparatus represent are considerable, and the recuperating energy in the lost steam enables an important economy to be realized in the products of a factory.

Nevertheless, there is one difficulty: the machines in question, being of intermittent functionment, the interrupted flow of steam from them has to be regulated, as otherwise the turbine could not be employed under these conditions.

The ingenious accumulator-regenerator of steam, designed by Mr. Rateau, has completely solved this problem. It transforms a discontinuous flow of steam into a continuous flow, whatever may be the prime motor.

Mr. Rateau has demonstrated* that not only the utilization of the exhaust steam was advantageous in the case when that steam escaped into the atmosphere, but even when it was condensed: there is an advantage in recuperating the energy of steam of atmospheric pressure down to the pressure in the condenser of a steam-turbine over effecting that recuperation in the large cylinder of a piston steam-engine.

Thus, not only is the use of the turbine rational in the cases of the pumping engines of mines, and of the engines of rolling mills, but it will be shown to be so in many steam-engines of multiple expansions.

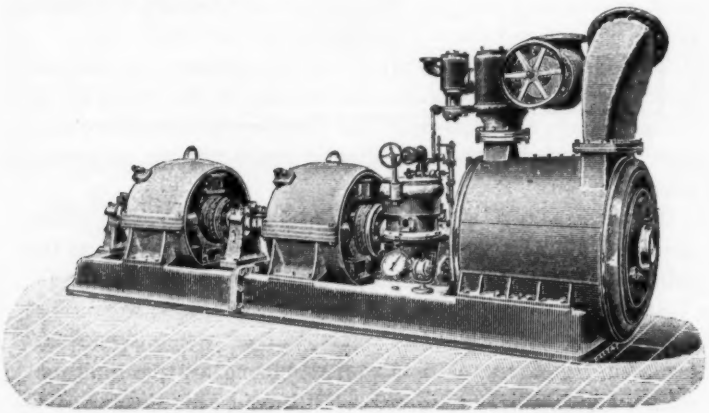


Fig. 27.—Low Pressure Turbo-Dynamo Group of 300 Horsepowers.

Figure 27 represents a low pressure turbo-dynamo group of 300 horsepowers erected in the mines of Bruay, in 1902. The turbine, of one body, uses the exhaust steam of a piston pumping engine therein, and expands it down to the condenser-pressure.

At the mean speed of 1,600 revolutions per minute, the tur-

* *Note on an accumulator-generator of steam*, by M. A. Rateau, Paris, 1902.
Utilisation des vapeurs d'echappement par l'emploi d'accumulators de vapeur et de turbines à condensation. Bulletin de la Société de l'Industrie Minérale, premiere livraison, 1903.
Utilisation des vapeurs d'echappement. Note publiée par la Société d'Exploitation des Appareils Rateau (accumulateurs de vapeur), 28 avenue de Suffren, Paris, Novembre, 1903.

bine works two dynamos placed at the end of the shaft, and is sufficient for the service of pit No. 5 of these mines.

When the pumping engine is not in use, steam is furnished to the turbine by the boiler, and is expanded from its entrance by means of an automatic expanding arrangement.

In Figure 28 are the curves of the economy of this turbo-dynamo in function of speed.

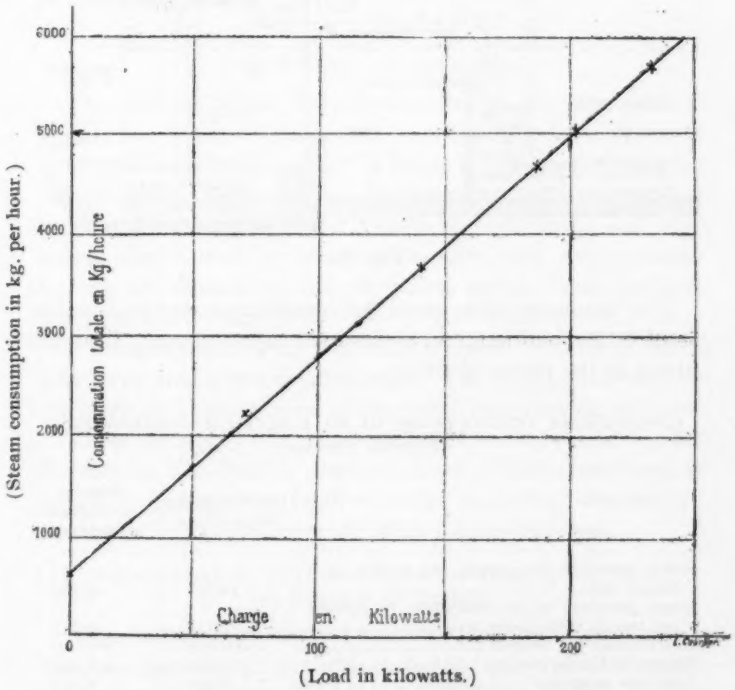


Fig. 28.—Low Pressure Turbo-Dynamo of 300 Horsepowers.

The figures which have served as basis for these curves, were obtained in the workshops of Sautter, Harlé and Co., by Messieurs Sauvage and Picou with every possible precaution.

The ensemble of the installation made in the mines of Bruay, is represented in Figure 29.

There is shown behind the turbine, the steam-accumulator which regulates the flow of the exhaust steam delivered by the pumping engine.

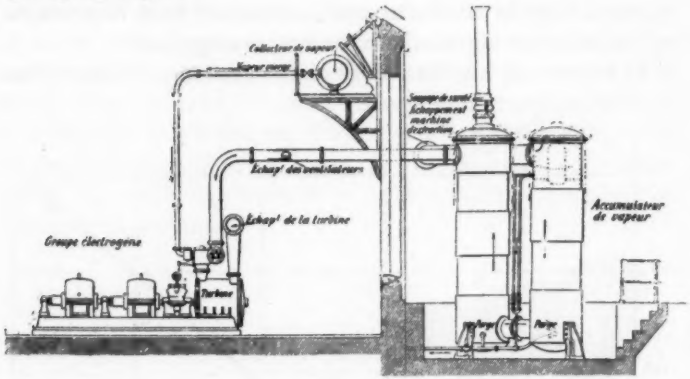


Fig. 29.

The following table gives the comparative results as calculated by the builders, and as obtained experimentally from the group in the mines of Bruay.

LOW-PRESSURE TURBO-DYNAMO OF 300 ELECTRICAL HORSEPOWERS.—RATEAU SYSTEM.

——	Calculated by the builders.	Results experimentally obtained.
Steam pressure, in pounds per square inch above zero	12.801	14.365
Steam pressure in the condenser, in pounds per square inch above zero	2.133	2.617
Revolutions per minute	1,800.000	1,800.000
Electrical horsepowers	300.000	336.000
Total net economy	0.570	0.580
Consumption of steam, in pounds weight per hour per electrical horsepower	38.360	37.368
Economy of the dynamos, measured by the method of opposition	.....	0.925

No more need be said regarding the innumerable applications that can be made of steam-turbines for the utilization of exhaust steam. There has been simply shown that the

employment of multicellular turbines allows the combination of machines able to utilize alternately or simultaneously two flows of steam; the one of low pressure, the other of high pressure, both in the best conditions of economy. A turbine could thus be supplied by the exhaust steam from a rolling mill or any other primary engine, or by live steam drawn from a boiler, and in any proportion whatever, and for any length of time desired.

V.—APPLICATIONS OF THE STEAM-TURBINE TO THE PROPULSION OF VESSELS.

The application of the steam-turbine to the propulsion of vessels has caused, during late years, numerous discussions.

Mr. Parsons has the honor of being the first who propelled vessels entirely by turbines. His first attempt, *The Turbinia*, was soon followed by other vessels of light hulls and high speeds constructed for the British navy, and, more recently, by the *Queen* and by the *Brighton*, packet boats between England and France.

It is impossible to here treat in a complete manner problems which required a long development. The purpose is to only set forth certain consequences of the experience thus far acquired.

These are stated in a communication recently made by Mr. Rateau to the British Institution of Naval Architects, in which he gives some inedited notes regarding the last experiments in propulsion made with Rateau turbines.

DIFFICULTIES IN THE APPLICATION OF TURBINES TO THE PROPULSION OF VESSELS.

Notwithstanding that, to a first view, the replacing of piston steam-engines by steam-turbines for the propulsion of vessels seems easy, and that the advantages of the change appears evident, a more profound study of the case discovers difficulties under the three following heads:

1st. The adjustment of the screw propellers to the great rotary speed of the turbines.

2d. The obtaining of a good economy when the power is reduced for reduced speeds of vessel.

3d. For performing the backing and maneuvering of the vessel.

ADJUSTMENT OF THE SCREWS TO THE GREAT ROTARY SPEED OF THE STEAM-TURBINE.

When the turbines are allowed to revolve at the speed which gives their maximum economy, they give excellent economic results, certainly better than can be obtained from the best piston steam-engines.

The figures that have been previously given for different applications of turbines, confirm this fact; as do, also, the published results of the experiments made with the turbines of Parsons.

Unfortunately, the rotary speed which assures a good economy with turbines, is too high for adaptation to helicoidal propellers, and the sacrifice, in part, of that economy becomes necessary if grave errors with the screws are to be avoided.

In order to turn the turbine slower than would be necessary with other applications of it, the number of working wheels must be increased, thus dividing the turbine into several parts, the steam cascading from the first part to the last part. On the other hand, in order to permit the screws to have a corresponding speed of rotation, they must be placed one, two or three on each of several shafts, and their surface must be so proportioned that their peripheral diameter is greater than their pitch.

These modifications diminish the economy of the screws, as well as they diminish the economy of the turbine; and there cannot be shown *a priori* that the mean economy of turbine and of screws is superior to that of a screw worked by a piston steam-engine.

The difficulties increase for slower speeds of vessel. This results, on the one part, from the fact that the total surface and, consequently, the dimensions of the screws, are determined by the greatest immersed transverse section of the vessel, while, on the other part, the size of the turbine is determined solely by the speed of its rotation, and not in the least respect by its power.

In measure as the speed of the vessel is reduced, the speed of the turbine must be proportionally reduced, so that the dimensions of the turbine increase either by the diameter of the wheels or by their number during the diminution of its power, which latter is approximately in the inverse ratio of the cubes of the speeds. There is then a limit of speed below which turbines cannot be practically employed.

Mr. Rateau has expressed the opinion that this limit is probably about 20 sea-miles per hour. For very large vessels and consequently, very large powers to be developed, this figure, perhaps, might be a little reduced; but there is a very small probability that it could be much lowered.

ECONOMY AT LOW SPEEDS.

If the steam-turbine is capable of giving good results as regards economy at maximum power, its economy at reduced speeds becomes, frankly, very bad, because of the diminution of the hydraulic economy as has previously been shown.

At reduced speed the expenditure of steam by a turbine is always more than by a piston steam-engine, but at normal speed this expenditure can be a great deal less.

This disadvantage would not be of much importance for such vessels as passenger-steamers which nearly always go at maximum speed, but, for war vessels, whose most economical speed is far less than their maximum speed, which indeed is only exceptionably employed, there results that their increased consumption due to the decreased economy, enormously diminishes their radius of action.

Parsons has proposed to remedy this defect by adding a supplementary turbine called the "cruising turbine" into which the steam is first introduced when the steaming is done at a low speed, nevertheless, this process does not improve the hydraulic economy, and the consumption of steam remains high.

The only efficacious way for obtaining an economic functionment when steaming at low speeds, is to employ, in conjunction with the turbine, a piston steam-engine of less power.

BACKING AND MANEUVERING.

With piston steam-engines, stopping and reversing are done in the simplest manner.

Turbines, on the contrary, are from their very nature irreversible.

Different inventors have proposed special arrangements of the bucketing allowing the same wheels to operate in both the forward and the backward direction. These attempts are not destined to succeed, because they can effect reversibility only by sacrificing the economy in the forward direction. Consequently, for backing, a piston steam-engine must be employed, or an additional steam-turbine.

But, unfortunately, the employment of a special turbine, has the disadvantage of greatly increasing the length of the space occupied by the machinery.

Mr. Rateau patented in 1898 the ingenious idea of placing in the interior of a principal turbine, and on its low pressure side, a backing turbine composed of several wheels with reversed bucketing, receiving steam from a special distributor.

This arrangement occupies less space than a separate turbine, and it is not detrimental when the vessel is going forward, as the backing wheels do not create any sensible resistance when the principal motor functions. It has been applied to the torpedo-boat *243* and to the *Libellule*.

But under any circumstances, it is difficult with turbines, to obtain a great backing speed. For that purpose, the engine must be able to develop as much power in backing as in going ahead; this is a serious disadvantage even with piston steam-engines and they are absolutely reversible.

As to the maneuvering in coming alongside of vessels, or of docks, the prompt stopping of the turbines has some difficulties:

Thus, when the steam-admission valve is closed, the screws continue to turn under the impulsion of the water, and easily entrain the wheels of the turbine, because their resistance to the movement of rotation is very feeble.

Also, the stopping and reversing, and, in general, the ease

and rapidity for maneuvering the vessel, are very serious difficulties with the steam-turbine, and especially for vessels of war.

APPLICATIONS THAT HAVE BEEN MADE TO VESSELS.

FRENCH TORPEDO BOAT 243.

In 1898 the French Navy began to experiment with steam-turbines for the propulsion of vessels of war.

That Navy ordered from the Forges et Chantiers de la Méditerranée the experimental torpedo boat *243* of 92 tons, furnished with Rateau turbines.

This vessel was provided with two screw-shafts on each of which was a turbine in which the steam was completely expanded from the pressure in the boiler to the pressure in the condenser.

Unfortunately, the programme adopted was to utilize the hull of an ordinary torpedo-boat, and of placing the turbines in the space habitually allotted to its piston steam-engine.

That programme caused much inconvenience and embarrassment in the installation of the turbines and screws, and it prevented the experiment from being as conclusive as it was intended to have been.

In fact, there is much difficulty in adapting turbines to a hull originally designed to receive the shaft of a single piston steam-engine. The form of the hull not being sufficiently flat in its lower part, there was an impossibility of lowering the shafts enough at the place where they left the turbines, so that these shafts had to be much inclined—about 11 degrees.

Each shaft carried three screws, and was held by brackets secured to the hull. These brackets had a considerable length, and a resistance that could not have been negligible.

Figure 23, Plate VI, shows the stern of the torpedo-boat with the screws.

The conditions of the installation were very unfavorable. The much inclination of the shafts rendered the economical performance of the screws quite defective, as was ascertained from various and prolonged experiments.

The turbines functioned in a very satisfactory manner, as stated in all the proces-verbaux of the experiments.

Below will be found copies of two of these proces-verbaux, in which are given the different speeds of vessel obtained with different pressures, and all the data necessary for valuing the qualities of the screws employed.

The two turbines are nominally of 900 horsepowers, and each is completely independent of the other. They contain at the exhaust end a single moving wheel for backing, which maneuver is consequently voluntarily sacrificed.

The speed of vessel obtained under the best conditions, has slightly exceeded 21 nautical miles per hour, while the speed which should have been obtained is about 24 nautical miles per hour, if the screws had given the same economy as those employed with the piston steam-engine at reduced speeds.

The screws employed with the turbine were much modified during the experiments. The number experimented with on each shaft was one, two and three, with very variable ratios of their pitch to their diameter, and having resistances proportionally variable.

The general conclusion that can be drawn from the experiments is the necessity of avoiding the inclination of the shafts, and the necessity of constructing the hull with proper adaptation for propulsion by turbines.

The turbine employed was of the kind first designed by the inventor, and had inside pillow-blocks. A flow of oil under pressure had been provided at the place where the shaft traverses the end of the turbine on its low pressure side.

An improper detail regarding this lubrication caused some trouble, the joint hindered the entrance of air, and entrained the oil into the condenser.

By the aid of a new system, entirely tight, it has been found possible to suppress even the low pressure pillow-block of the turbine itself, and to completely correct the trouble stated.

These details are not useless, because of many rumors concerning the prolonged experiments with the torpedo-boat *243*, attributing the speed failure to the bad functioning of the turbines.

The experiments have been extended through the years 1902 and 1903, and will soon be completed. The French Navy has desired to ascertain as completely as possible the difficulties of the problem of screws worked at great rotary speeds, and the means of overcoming them.

Up to the present date, not less than seven systems of screws have been experimented with, designed by different specialists, and combined in various groupings.

Figure 24 Plate VI, represents the torpedo boat *243* under way.

LIBELLULE.

Another vessel of less tonnage—a vedette-torpedo boat—has been constructed for the French Navy with its turbine in two bodies instead of in one body as the turbine of *243* was, but with analogous power.

This vessel has only a single shaft. The experiments with it have not yet been commenced. The boilers are of a new system, and will be tried for the first time in the *Libellule;* the experiments with them are not yet completed.

BOAT BY YARROW AND COMPANY.

Messieurs Yarrow and Company have recently constructed, with a Rateau turbine, a light boat of the same type as the *Tarantula* built by the same company and furnished with a Parsons turbine.

This type of boat with the exception of the screws, is identical with the first class torpedo-boats of the British Navy. Its displacement is 140 tons and its length is 151 feet.

Its boilers, of the well known Yarrow type, have given the vessel by means of a piston steam-engine, a speed of about 27 geographical miles per hour.

This vessel is furnished with three shafts each carrying screws. The central shaft is worked by a piston steam-engine of 250 effective horsepowers; and this engine is entirely independent of the turbines.

The side shafts are arranged to receive for each of them one or two screws, and are each operated by a turbine in two bodies or in "cascade."

FRENCH TORPEDO-BOAT No. 248.

Draught of Water { Forward, 3.6418 feet. Aft, 4.4948 feet. Mean, 4.0683 feet. Difference, 0.8530 feet.

Screws *F C M* { Number of blades in each screw, 4. Diameter of the screw, 1.96854 feet. Pitch of the screw, 1.63750 feet. Fraction of the pitch per blade, 0.11000.

EXPERIMENTS ON THE CONSUMPTION AT 20 KNOTS MADE JANUARY 22, 1903.

	Times of observation.	Direction of the base.	Time of making the run, in minutes and seconds.	Number of revolutions. By the counter. Starboard.	Port.	Means per minute.	Speed in knots.	Advance of vessel in feet per revolution.	Observed pressures by manometers, in pounds per square inch above the atmosphere. At the pumping plant.	Boilers. Forward.	Aft.	Collector of the steam.	Turbines. Starboard.	Port.	Pressure in condenser in pounds per square inch above zero.	Feed water heater.	Air pressure in boiler room, in pounds per square inch above the atmosphere.	Revolutions of the blowing engines per minute.	Fraction of the steam valve opening used. [Steam throttling at the turbines.]	Temperatures in Fahr. deg. Feed water.	Engine room.	Revolutions per minute of the steam engines driving the pumping plant.
	2 h. 17 m.	Beginning.	...	...	...	...	...	216.2	209.1	211.9	209.1	133.7	126.6	1.354	0.013	1.741	260	0.375	201.2	80.6	66
a	2 h. 30 m.	N.S. 3′-13″	5,490	5,600	1,723.83	20.438	1.199603	217.6	211.9	213.3	210.5	133.7	126.6	1.354	0.015	1.741	275	0.375	206.6	82.4	66
b	2 h. 42 m.	S.N. 3′- 3″	5.320	5,450	1,765.57	21.554	1.235265	219.0	213.3	214.8	213.3	135.1	128.0	1.354	0.015	1.741	265	0.375	204 8	80.6	68
c	2 h. 56 m.	N.S. 3′-15″	5,590	5,700	1,738.46	20.228	1.177326	217.3	210.5	213.3	210.5	130.9	123.7	1.335	0.015	1.934	240	0.375	203.0	82.4	68
I		Means $= \frac{a+b+b+c}{4} =$			1,748.36	20.9435	1.211873	Slip of screws = 0.260.														
	3 h. 17 m.	Free route.	...	...	...	...	...	216.2	210.5	211 9	209.1	133.7	126.6	1.335	0.013	1.837	250	0.375	204 8	80.6	66
d	3 h. 30 m.	S.N. 2′-56½″	5,150	5,290	1,774.50	22.349	1.273421	230.4	221.9	226.1	220.5	135.1	128.0	1.335	0.015	1.934	270	0.375	204.8	82.4	67
e	3 h. 42 m.	N.S. 3′-17″	5,740	5,860	1,766.50	20.023	1.146880	229.0	220.5	223.3	219.0	133.7	126.6	1.335	0.015	1.934	265	0.375	206.6	82.4	69
f	3 h. 54 m.	S.N. 2′-54″	5,140	5,250	1,791 38	22.669	1.280442	227.6	221.9	223.3	214.8	138.0	130.8	1.335	0.015	1.934	265	0.375	206.6	80.6	67
II		Means $= \frac{d+e+e+f}{4} =$			1,774.72	21 266	1.212135	Slip of screws = 0.260.														
		Means $= \frac{I+II}{2} =$			1,761.54	21.105	1.212004	Slip of screws = 0.260.														
	4 h. 17 m.	End.	...	...	...	...	...	227.6	221.9	223.3	220.5	133.7	126.6	1.373	0.015	1.934	265	0.375	204.8	80.6	66

OBSERVATIONS.—Total grate surface, 44 9 square feet. Duration of experiment, 2 hours. *Consumption of Coal:* Total pounds, 7,231.0; pounds per hour, 3,615.5; pounds per hour per square foot of grate, 80.523; pounds per hour per mile (20.972), 142.4. *Consumption of Oil:* Total olive oil, in pounds, 39.68; total mineral oil, in pounds, 13.22; aggregate pounds, 52.90. The general functionment has been very satisfactory.

Total number of revolutions during the experiment, starboard, 208,100; port, 212,740.

Mean number of revolutions made per minute, starboard, 1,734.17; port, 1,772.83.

Mean number of revolutions made per minute for the two turbines, 1,753.50.

Mean advance of the vessel in feet per revolution = 1.212004.

Speed of the vessel in knots $= \frac{1,753.5 \times 1.212004}{101.33664} = 20.972.$

FRENCH TORPEDO-BOAT No. 243.

Draught of water: Forward, 3.6909 feet. Aft, 4.4445 feet. Mean, 4.0677 feet. Difference, 0.7536 feet.

Screws *F C M* modified: Number of blades in each screw, 4. Diameter of the screw, 1.96854 feet. Pitch of the screw, 1.63943 feet. Fraction of the pitch per blade, 0.11060.

EXPERIMENTS MADE SEPTEMBER 17, 1903. (SCREWS *AV* AND *M* IN PLACE.)

Times of observation.	Direction of the base.	Time of making the run, in minutes and seconds.	Number of revolutions. By the counter. Starboard.	Number of revolutions. By the counter. Port.	Number of revolutions. Means per minute.	Speed in knots.	Advance of vessel in feet per revolution.	Observed pressures by manometers, in pounds per square inch above the atmosphere. At the pumping plant.	Boilers. Forward.	Boilers. Aft.	Collector of the steam.	Turbines, mean of starboard and port.	Pressure in condenser in pounds per square inch above zero.	Feed water heater.	Air pressure in boiler room, in pounds per square inch above atmosphere.	Revolutions of the blowing engines per minute.	Fractions of the steam valve opening used. [Steam throttling at the turbines.]	Temperature in Fahrenheit degrees of the feed water.	Revolutions per minute of the steam-engines driving the pumping plant.
2 h. 57 m.	DA	5′-48″	6,870	7,100	1,204.3	14.054	1.181755	210.5	196.3	203.4	200.5	42.67	1.315	−5.80	0.484	242	0.125	188.6	40
3 h. 12 m.	AD	5′-29″	6,610	6,820	1,224.6	14.866	1.229327	207.7	197.7	200.5	197.7	41.25	1.325	−3.87	0.387	226	0.125	188.6	40
		Means,			1,214.4	14.460	1.205541	Slip of screws = 0.2640.											
3 h. 30 m.	DA	4′-54″	7,000	7,300	1,459.1	16.635	1.154524	206.3	199.1	199.1	199.1	69.69	1.325	+1.93	0.677	240	0.250	201.2	40
3 h. 40 m.	AD	4′-54″	7,090	7,410	1,479.5	16.635	1.138448	210.5	199.1	203.4	202.0	72.54	1.325	+1.93	0.677	242	0.250	202.1	40
		Means,			1,469.3	16.635	1.146486	Slip of screws = 0.3006.											
3 h. 56 m.	DA	4′-21″	7,300	7,760	1,731.0	18.739	1.096125	207.7	199.1	200.5	199.1	99.56	1.315	+7.74	1.161	250	0.333	208.9	52
4 h. 8 m.	AD	4′-31″	7,680	8,160	1,753.5	18.047	1.042320	214.8	206.3	207.7	206.3	100.98	1.315	+7.74	1.161	252	0.333	207.5	56
4 h. 20 m.	DA	4′-15″	7,230	7,680	1,754.1	19.180	1.107280	214.8	206.3	207.7	203.4	102.40	1.315	+7.74	1.354	252	0.333	210.2	56
		Means,			1,748.0	18 803	1.072011	Slip of screws = 0 3462.											
4 h. 33 m.	AD	4′-18″	8,310	8,760	1,984.9	18.957	0.967189	236.1	227.6	229.0	226.1	128.00	1.296	+15.44	1.741	280	0.375	213.8	60
4 h. 43 m.	DA	3′-55″	7,630	8,000	1,995.3	20.812	1.056099	227.6	220 5	223.3	217.6	124.45	1.296	+13.54	1.934	280	0.375	212.0	60
4 h. 55 m.	AD	4′-18″	8,480	8,940	2,025.5	18.957	0.947504	226.1	219.0	216.2	214.8	126.58	1.296	+15.44	1.741	280	0.375	210.2	60
		Means,			2,000.2	19 8845	1.006723	Slip of screws = 0.3860.											

Observations.—Started blowers at 1 h. 55 m. Equalized the turbines at 2 h. 10 m. Began the experiments at 2 h. 30 m. Ended the experiments at 5 h. 20 m. Smooth water. Light breeze from E. to S. Barometer 30.31 inches. Consumed 28 quarts of olive oil. Consumed 4 quarts of mineral oil. The general functionment has been very satisfactory.

The combination of a piston steam-engine and steam-turbines adopted by Messieurs Yarrow & Co., had been previously suggested by Mr. Soliani, director of the Ansaldo workshops. The combination can be made in various ways.

Mr. Rateau has shown a way of combining the piston steam-engine with steam-turbines, by directing its exhaust steam into either the high pressure or into the low pressure turbine, according to the case.

Without entering into details, the results of the experiments show that the turbines make 2,000 revolutions per minute (they were calculated for 1,500 revolutions), and that they develop over 2,000 horsepowers with a consumption of a little more than 8.8 pounds of steam per hour per horsepower.

The maximum speed attained during the experiments which have been made since October last, is 26.25 geographical miles per hour. Several different screws have been tried with one screw and two screws on each shaft.

The opinion of Mr. Yarrow's engineers is, that it would be possible to attain, if not the speed of 27 geographical miles per hour, at least a speed not much inferior.

As regards the mechanical or practical working, the Rateau turbine in the Yarrow boat showed itself an excellent machine, as did also the Rateau turbine in *243*.

There was no accident of any kind, neither rupture of a blade, nor interference of mechanism, nor vibrations, and the admission is now generally made that the multicellular turbine is not inferior in any respect to any other system known on board of vessels.

Figure 25 Plate VI, gives a general view of the vessel constructed by Messrs. Yarrow & Co.

RESUMÉ AND CONCLUSIONS.

To sum up: there has been given in this work a general classification of the different systems of steam-turbines, a classification in which has been included not only the systems actually known to, and employed in, practice; but, also, those which could be formed by the application of the general principles of mechanics to this kind of machine.

There has been pointed out the place which the Rateau turbine occupies in such a classification, and from it the immediate result of the advantages of the multicellular system of action.

The question of economy has then been investigated by analyzing the different elements on which it depends, that is to say, first, the hydraulic economy; the losses occasioned by the leakage of steam, and by its frictions and eddies.

Then, this analysis has been applied to the comparison of the two turbines of action, namely, the multicellular turbine with pressures in stages, and the turbine having groups of wheels with velocities in stages.

This discussion has brought forward in a complete manner the economic superiority of the first combination.

Divers combinations recently announced as new systems have been examined, and their economic and mechanical performances determined.

The purpose of the second part of this work has been the exposition of the applications which have been made of the Rateau turbine; first, to the generation of electricity; and, afterwards, to the turbo-pumps and to the turbo-ventilators; then, to the utilization of exhaust steam; and, finally, to the propulsion of vessels.

It is hoped that this review, sometimes summary, will suffice to show the general interest that attaches to the system invented by Mr. Rateau, an interest so much the greater as that engineer has known how to create the necessary methods for calculating the turbines in all the various conditions of their application with a rigor proved by the preceding examples, and which has not been attained by even the piston steam-engines.

In conclusion, the remark seems proper that if the honor of the first industrial turbine belongs to England in the person of Mr. Parsons, the labors of Mr. Rateau and of the firm of Sautter, Harlé and Company, assures, nevertheless, on their part for France, a marked place in the development of the new steam-motor.

TORPEDO BOAT No. 20, *GOLDSBOROUGH.*

DESCRIPTION AND OFFICIAL TRIAL PERFORMANCE.

By G. H. Thayer, Draughtsman, U. S. N.

The *Goldsborough* (built by Wolff and Zwicker Iron Works, Portland, Oregon) was one of three torpedo boats authorized by the Act of March 3, 1897.

The contract was signed July 30, 1897, the price being $214,500 (exclusive of ordnance and equipment fittings, which were supplied by the Government). The time allowance for completion was eighteen months.

The speed guaranteed in the original contract was 30 knots per hour, to be maintained for two consecutive hours over a course not less than thirty nautical miles in length. The vessel to carry the following weights additional to that of the complete hull and machinery, with spare parts and the water necessary for the trial, viz: Torpedo tubes, gear and torpedoes, rapid-fire guns and ammunition, 12.6 tons; coal, 20 tons; crew, boats, anchors, electric plant, equipment, etc., 9 tons.

After a long series of trials, accidents and delays, the Navy Department (January 21, 1902) declared the contract forfeited on the part of the contractors. On March 3, 1905, an Act of Congress was approved authorizing the Secretary of the Navy to accept the *Stringham* and *Goldsborough* after a satisfactory trial at reduced speed. It was decided to run the standardizing trial up to and including 27 knots, and the one-hour's run not to exceed 27 knots.

HULL DATA.

Length between perpendiculars, feet and inches	198- 0
L.W.L., feet and inches	198- 0
Beam, extreme, feet and inches	20- 7¼
L.W.L., feet and inches	19-11

Depth, M.S., feet and inches	12- 6
Mean draught on trial, feet and inches	6-10 5/16
Displacement on trial, tons	255.5
full load, tons	360
per inch, L.W.L., tons	6.3
Area of immersed M.S., square feet	76
L.W. plane, square feet	2,670
Center of gravity of L.W. plane aft of M.S., feet and inches	7- 9
buoyancy above keel, feet	3.6
Transverse metacenter above center of buoyancy, feet and inches	6-11
Longitudinal metacenter above center of buoyancy, feet	595
Number of frames	117
Watertight compartments	22

HULL CONSTRUCTION.

The hull is constructed of mild steel of domestic manufacture and galvanized to two feet above the load water line.

The frames are spaced 20 inches between centers and are 2¼ inches × 1½ inches × 2.3 pounds angle steel amidships and 2 inches × 1⅜ inches × 2.1 pounds angle steel at ends of ship.

Reverse frames are 2 inches × 1⅜ inches × 2.1 pounds angle steel. There are eight belt frames, 6 inches deep, in engine space.

The flat keel is 10-pound plate amidship, reduced to 9-pound and 8-pound plates at end.

The center-line keelson is 7½-pound plate in engine and boiler rooms, and intercostal 5-pound plate at ends.

The shell plating is 6-pound plate amidship for half the length of the ship, reduced to 4.43 pounds at ends, except garboard and sheer strakes, which are—garboard, 8-pound to 6½-pound; sheer strake, 8½-pound to 5-pound.

The watertight bulkheads, of which there are eleven, extending to the deck, divide the ship into twelve main compartments. The forepeak, storeroom and capstan room, machinists' room, with locker, galley, athwartship coal bunker, forward boiler room, after boiler room, engine room, crew's quarters, officers' quarters, officers' toilet and pantry and storeroom in the stern.

Magazine and storerooms are provided below crew's and machinists' quarters.

The bridge is located on top of the forward conning tower, and is fitted with the steering wheel controlling the steering engine, which is located in the engine room; a tell-tale, indicating the position of the tiller, engine-room indicators and voice tube to engine room.

The forward conning tower, which is of 7½-pound steel plate, contains steering wheel, voice tube to engine room, chart locker and entrance hatch to machinists' quarters.

The after conning tower, which is also of 7½-pound steel plate, contains a hand steering wheel, engine-room indicators and entrance hatch to crew's quarters.

The boats, two of which are 20 feet long, of Navy model, and one of the canvas folding type, are carried on deck amidships.

Two anchors of the Baldt stockless type are carried, each stowed on skids, on the sides of turtle deck. These anchors weigh about 625 pounds each.

ARMAMENT.

The battery consists of four 6-pounder semi-automatic rapid-fire guns, mounted on pedestal mounts with hydraulic recoils. These guns are located as follows: One on the starboard side, over the forward fireroom coal bunker; one on the port side, over the after fireroom coal bunker; one on the starboard side, over the crew's quarters; one on the port side over the officers' quarters. This location gives all the guns the maximum amount of train combined with efficiency.

Torpedo tubes, of which there are two, located on the center line of the ship, train on either beam. The forward tube is located over the after part of the engine room and the after tube is located aft on the stern and over the after part of the officers' quarters. Four modern Whitehead torpedoes of the latest pattern are carried, two in the tubes and two under the turtleback forward.

MACHINERY.

Main Engines.—There are two triple-expansion, four-cylinder engines, of the vertical, inverted type, placed back to back in the same engine room, and designed to develop a total of

6,000 horsepower at 360 revolutions per minute. The engines are right and left, turn outboard, and drive twin screws.

The arrangement of the cylinders is: High pressure, intermediate pressure, forward low pressure and after low pressure. The cranks are at 90 degrees, the H.P. and I.P. cranks being opposite, as are also the 1st L.P. and 2d L.P. cranks.

The H.P. and I.P. cylinders are in one casting, as is also the case with the two L.P. cylinders; the two castings are bolted together and supported on a framing of nickel-steel vertical columns well stayed with diagonal braces.

The cylinder covers are composition, cast with stiffening ribs, and contain the steam ports and passages; also the valve liners.

The main valves are all single-ported plug valves, of the piston type, one for the H.P. cylinder and two each for the I.P., 1st L.P. and 2d L.P. cylinders; these valves are actuated (through a rock shaft) by means of a valve gear of the Stephenson link type, with double bars. The link block is carried at the center of a vertical link, the lower end of which connects to an arm of the rock shaft, and the upper end to a vibrating lever hung from the cylinder cover. The rock shaft turns in bearings carried by the back frames and main guide castings, and operates the main valves by means of arms forged from the solid. Each valve stem and its crosshead is in one forging of nickel-steel. The eccentrics are of cast steel, in two parts, bolted together and keyed to raised seatings on the crank shafts. The eccentric straps are of composition, lined with white metal, and the eccentric rods are of forged steel. Adjustment of cut-off is effected by means of a sliding block in the reverse-shaft arm which is shifted by means of a threaded bolt.

CYLINDER DATA.	H.P.	M.P.	Each. (2) L.P.
Diameter of cylinders, starboard, inches	19¾	31¾	35¾
port, inches	19¾	31¾	35¾
Liners in H.P. and I.P.	—	—	—
Stroke, inches	20	20	20
Thickness of bonnets, starboard, inch	⅞	⅞	⅝
port, inch	⅝	⅝	⅝

	H.P.	M.P	Each. (2) L.P.
Number of valves	1	2	2
Diameter of valves, inches	12 1/8	10 1/8	10 1/8
valve stem, inches	1 1/2	1 1/2	1 1/2
Side of valve on which steam is taken	inside	outside	1st inside 2d outside
Valve travel, inches	4 3/4	5 1/4	5 1/4
Steam lead, top, inch	1/16	1/16	1/32
bottom, inch	1/32	3/32	1/4
Cut-off, top, inches	14 1/4	14 1/4	14 1/4
bottom, inches	13 1/4	13	13 1/4
Exhaust, top, inches	17 1/4	17 1/8	17 1/8
bottom, inches	17 3/8	17 1/4	17 1/4

The main pistons are conical, of cast steel, and fitted with two hard cast-iron snap rings. Each piston rod is hollow and, with its crosshead and slipper, is in one piece forged of nickel-steel and oil tempered, the slipper being faced with white metal. The piston rods are secured to the piston by means of a taper fit and a nut.

PISTON RODS.

	H.P.	I.P.	Each. L.P.
Diameter of piston rod, inches	4	4	4
axial hole, inches	2 1/2	2 1/2	2 1/8
Width of crosshead slipper, inches	12	12	10
Length of crosshead slipper, inches	14	14	14
Width (total) backing surface, inches	5 1/2	5 1/2	5 1/2

The crosshead guides, of the slipper type, are of cast iron, hollow, for water circulation. The upper ends are bolted to lugs cast on the lower cylinder covers, and the lower ends bolted to forged-steel strongbacks, which are, in turn, secured to the engine columns. The backing guides are of cast iron and bolted to the face of the go-ahead guides.

The connecting rods are hollow. They are of forged nickel-steel, oil-tempered. The crosshead pins are hollow, of oil-tempered nickel-steel, and are shrunk into the eyes of the connecting rods. The hollow space which is used as an oil reservoir, communicates by means of a small hole with the upper surface of the pin. The crankpin caps are of composition, and they, with the lower end of the connecting rods, are lined with white metal. All crank-pin and crosshead-pin bolts are of nickel-steel.

CONNECTING RODS.

Length of connecting rods, center to center, inches	44
Diameter at neck, inches	3⅝
bottom end, inches	4¼
of axial hole, inches	2
crosshead pin, inches	5
Length of crosshead pin, inches	7
Diameter of axial hole, inches	2½
crosshead-pin bolt, inches	2
crank-pin bolts, inches	2½

Each crank shaft is in one section, hollow, of forged nickel-steel, oil-tempered. The cranks are at 90 degrees. The axial hole in the crank pin is utilized as an oil reservoir, being connected by a small hole to a centrifugal oiler, fastened to the side of the crank web, and having a small hole communicating with the outside surface of the crank pin, through which the oil is thrown by centrifugal force.

CRANK-SHAFT DATA.

Crank pin, diameter, inches	7½
length, inches	10
Diameter axial hole, inches	3½
of crank shaft, inches	7½
axial hole, inches	3½
eccentric seating, inches	7⅝
coupling flange, inches	13½
Thickness of coupling flange, inches	1½
Diameter of coupling bolts, inches	1⅝½
Crank webs, width, inches	8½ and 7⅝
thickness, inches	4
Diameter of main bearings, inches	7½
Length of main bearings, one H.P., I.P., inches	8
four H.P., I.P., inches	12
Diameter of main bearing bolts (2), inches	2

The engine bed consists of five main-bearing bridges, bolted to two steel angle bars by body-bound bolts. These angles are seated on and bolted by fitted bolts to the main-engine keelsons. Each bridge has an extension or arm reaching down, through the end of which a longitudinal bolt passes under the engine, clearing the cranks and binding all the bridges together, thus overcoming the tendency to roll or twist laterally. The main-bearing bottom brasses are circular, to facilitate re-

moval, and the caps are hollow, for water circulation; they are lined with white metal.

The thrust shaft, of oil-tempered nickel-steel, is in a short section connected to the crank and line shafts by means of flanges and taper bolts. It works in a thrust bearing of the ordinary horseshoe type, the body of the bearing is of composition so shaped as to form an oil reservoir, and has side lugs or brackets for bolting it between the main-engine foundation angles.

The horseshoes are of composition faced with white metal. Oil ways run to the working faces from a reservoir cored in the top and provided with wick feeds. The shoes are held in place by extension lugs fitting between lock nuts threaded on side adjusting rods extending the length of the box.

THRUST-SHAFT DATA.

Diameter of flange couplings, inches	13½
shaft, inches	7
axial hole, inches	4½
over collars, inches	11
Number of collars	7
Thickness of collars, inches	1⅛
Space between collars, inches	2½
Number of horseshoes	7
Diameter of side adjusting rods, inches	2
Length of thrust-shaft spring bearings, inches	7½
Diameter of thrust-shaft spring bearings, inches	7½

The line and propeller shafts are also of oil-tempered nickel-steel. The line shafting is connected to the thrust shaft by means of flanges, and to the propeller shaft by means of a taper and socket joint with cross key and feather. The shaft is covered with a sleeve where it passes through the stern tube. The after end of the propeller shaft is turned with a taper to fit the propeller boss.

LINE-SHAFT DATA.

Diameter of shaft, inches	7
axial hole, inches	4½
spring bearing, inches	7½
Length of spring bearing, inches	12½
Diameter over sleeve, inches	7½

Diameter in stuffing box, inches	7¾
Length of strut bearing, inches	15¾
Diameter of strut bearing, inches	7¾
Coupling sockets, outside diameter, inches	10¼
taper, per foot, inch	¾
Length of taper, inches	13¾
Cross keys, inches	3¼×1
Feather key, inches	1¼× ¾

The propellers are three-bladed, cast solid, of manganese-bronze and polished. Each propeller is fitted to the taper on its shaft and secured by means of a feather key and a nut on the end of the shaft, and locked in place. The nut is threaded in the opposite direction to the rotation of the propeller when going ahead.

PROPELLER DATA.

Number of blades	3
Diameter, feet and inches	7– 6
Mean pitch, starboard, feet and inches	10– 0.18
port, feet and inches	10– 4.8
Ratio, pitch ÷ diameter	1.36
Helicoidal area, square feet	37
Projected area, square feet	29.04
Disc area, square feet	44.179
Ratio, projected area ÷ disc area	.656
Area of immersed midship section	78.4
Ratio disc area to area immersed midship section	.563
projected area to area immersed midship section	.37
Diameter of hub, inches	14
nut thread, inches	3

REVERSING ENGINE.

The reversing engine consists of an oscillating cylinder, with the trunnion plate bolted to the vertical web of the foundation angle bar between the I.P. and 1st L.P. engines.

The end of the piston rod is directly connected to an arm forged on the weight shaft. The valve of the reversing engine is of the common "D" type, actuated by the ordinary floating lever system. A hollow shaft extends to an operating hand lever, located near the main throttle valve at the front of the engine. Through this hollow shaft runs an extension stem to the steam valve, thus giving the machinist in charge admirable facilities for handling his machinery. The steam and

exhaust pipes of the reversing engine are connected through the trunnions. At the forward end of the engine is also located a hand wheel, which can be thrown into gear with the weight shaft, to reverse the engine in case of an accident to the steam gear.

REVERSING-ENGINE DATA.

Diameter of steam cylinder, inches	4
Stroke of piston, inches	9½
Diameter of piston rod, inches	1¼
connecting pin, inches	1⅝
weight shaft, inches	3½
hand reverse wheel, inches	24
Number of turns of wheel to reverse	1¼

The turning gear, at the aft end of the engine, consists of a worm turned by a ratchet lever that can be thrown in mesh with a worm gear keyed on the flanges connecting the crank and thrust shafts.

TURNING-GEAR DATA.

Number of teeth in worm wheel	58
Pitch of worm, inches	1.245
Length of ratchet handle, inches	24

The condenser, into which both engines exhaust, is located on the center line of the ship, between the engines. The shell, which is of circular section, is formed of copper stiffened by circumferential corrugations. The condenser is curved in a vertical plane, the tubes being bent to concentric curvature and the ends expanded in the composition tube sheets, thus dispensing with ferrules and packing, the curvature of the tubes allowing for expansion from heat. The water chests are formed of sheet copper, the after one containing the propeller, by means of which water is drawn or forced through the tubes when the ship is at rest or under slow speed. There are two openings through the plate, with strainers and scoops, to force the water through the condenser when the vessel is under way.

CONDENSER DATA.

Diameter of shell, inside, feet and inches	3- 6
Thickness of shell	B.W.G. No. 10.
Number of tubes	1,972

Thickness of tubes	B.W.G. No. 18.
Average length of tubes, feet and inches	12- 9
Spacing of tubes, inch	$1\frac{1}{16}$
Cooling surface, square feet	4,006.66
Ratio of cooling surface to I.H.P.	.67
Diameter of exhaust pipe (2), inches	$18\frac{1}{2}$
injection pipe, inches	18
air-pump suction (2), inches	7

The main air pumps are of the trunk-bucket type, with floating top and foot valve plate, and are located at the forward end of the engines, under the floor plates, hanging by side brackets to the main-engine foundation angles. The air pumps are driven by a crank shaft having a projection at the after end that dovetails into a recess in the forward end of the main-engine crank shaft, thus allowing a slight independent alignment of the two shafts.

MAIN AIR-PUMP DATA.

Diameter of bucket, inches	$20\frac{1}{2}$
Stroke of bucket, inches	$3\frac{1}{2}$
Diameter of trunk, inches	$6\frac{1}{4}$
Number of valves in bucket	20
foot plate	20
floating top	25
Diameter of valves, inches	$2\frac{1}{2}$
Lift of valves, inch	$\frac{1}{4}$
Length of connecting rod, inches	$13\frac{1}{2}$
Diameter of crank shaft, inches	$3\frac{1}{2}$
Trunk pin, inches	$2\frac{1}{2} \times 5\frac{1}{2}$
Crank pin, inches	$3\frac{1}{2} \times 4$
Crank-shaft bearings, inches	$3\frac{1}{2} \times 5\frac{1}{2}$
Suction pipe diameter, inches	7
Discharge pipe diameter, inches	7

The main circulating pump consists of a propeller, located in the after water way of the condenser, driven by an independent, vertical, inverted, single-cylinder, slide-valve engine. The cylinder of this engine is supported on light, turned, wrought-steel columns. The forward main bearing forms the thrust, the collars on the shaft running incased in the white metal of the bearing. The engine is reversed by means of a shifting eccentric.

CIRCULATING-PUMP DATA.

Diameter of cylinder, inches	6
Stroke of piston, inches	5
Diameter of piston rod, inches	$1\frac{1}{2}$
Crosshead pin, inches	$1\frac{5}{8} \times 4\frac{3}{8}$
Axial hole in pin, inch	1
Crosshead slipper, inches	$3\frac{1}{4} \times 3\frac{7}{16}$
Length of connecting rod (center to center), inches	10
Crank pin, inches	$2 \times 3\frac{1}{4}$
Axial hole in pin, inches	$1\frac{1}{2}$
Crank webs, inches	$1 \times 2\frac{1}{4}$
shafts, diameter, inches	$1\frac{3}{4}$
shaft bearings, inches	$2\frac{1}{4} \times 1\frac{3}{4}$
thrust bearings, inches	$4\frac{1}{4} \times 5\frac{1}{2}$
Number of rings	2
Circulating propeller, diameter, inches	20
pitch, inches	18
revolutions per minute	300

STEAM PIPES.

Two main steam pipes, one on each side, lead aft to the engines, with a cross connection and stop valve between them forward of the engines. There is also a main stop valve on each main line at the engine-room bulkhead. The throttle valves are of the rotating type geared to hand wheels, with divisions on them to indicate the degree of opening.

The auxiliary steam pipe pierces the bulkhead on the starboard side, has a cross connection to the main steam line, and a branch leading to the by-pass valve connected to the L.P. receivers.

BOILERS.

There are three water-tube boilers of the Thornycroft "Daring" type, one of which is located in the forward boiler room and two in the after boiler room. Each boiler has two furnaces, with two fire doors to a furnace. The furnaces are lined with firebrick. The grate bars are of mild steel made up in sections of five bars to each section. The boiler casing is of thin galvanized iron, lined with asbestos millboard. There is one twin-spring safety valve and one sentinel valve to each boiler. The feed water discharges into the lower part of the steam drum through a Thornycroft feed-water

regulator connected to each boiler. The boiler in the forward boiler room and the forward boiler in the after boiler room discharge their smoke into the same smoke pipe, which is located over the watertight bulkhead between them. The after boiler in the after boiler room has its own smoke pipe located forward of the engine-room hatch.

BOILER DATA.

Number of boilers	3
Steam pressure, pounds	250
Length over casings, feet and inches	10- 2
Width over casings, feet and inches	13- 7
Height ash pan to top of casing, feet and inches	10- 3
Steam drums, length, feet and inches	12-11$\frac{1}{4}$
drum, diameter, inches	37
thickness, inch	$\frac{7}{8}$
Water drums, number for each boiler	3
Side drums (2), diameter, inches	6$\frac{1}{2}$
thickness, inch	$\frac{1}{2}$
Center drum, diameter, inches	18
thickness, inch	$\frac{5}{8}$
Tubes, downflow, diameter, inches	3$\frac{1}{2}$
thickness, inch	$\frac{9}{16}$
number to each boiler	14
generating, diameter, O. S. inches	1$\frac{1}{2}$ and 1$\frac{1}{4}$
thickness	No. 11, B.W.G.
number to each boiler	1,780
Baffle tubes, diameter, inches	1$\frac{1}{2}$ × 1$\frac{1}{4}$
thickness	No. 11, B.W.G.
number to one boiler	808
Furnaces, number to each boiler	2
doors to each boiler	4
width of grate, feet and inches	4- 9$\frac{1}{2}$
length of grate, feet and inches	7- 6
Total heating surface of three boilers, square feet	13,500
grate surface of three boilers, square feet	216
Ratio grate surface to heating surface	1 : 62.5
Smoke pipes, height above grates, feet and inches	23- 6
area of forward pipe, square inches	2,501.82
after pipe, square inches	1,320.25
Ratio of stack area to G. S.	17.7 : 1
I.H.P. per square foot of grate surface, main engines	27.8

FORCED DRAFT.

The closed-fireroom system of forced draft is used, air being supplied by three fan blowers, built by the contractors from

their own designs. The blower engines are direct-connected, of the two-cylinder, horizontal, opposed type, with cranks at 180 degrees, and piston valves. The working parts are enclosed in a sheet-iron casing to protect them from the dirt and coal dust.

One blower is located in the forward fireroom, on a shelf fastened to the forward athwartship bulkhead, and takes its air from an air box, over which is mounted the forward 48-inch copper cowl. In the after fireroom are located the other two blowers, on shelves fastened to the starboard and port coal-bunker bulkheads, and they take their air from an air box between them, over which is mounted the after 72-inch copper cowl.

BLOWER ENGINE DATA.

Diameter of cylinders, inches	$3\frac{1}{4}$
Stroke of pistons, inches	$3\frac{1}{4}$
Diameter of piston rods, inch	$\frac{5}{8}$
valves, inches	$1\frac{5}{16}$
Travel of valves, inches	1
Crosshead pins, inches	$1\frac{1}{8} \times 4\frac{5}{8}$
slippers, inches	$2\frac{3}{4} \times 4$
Length of connecting rod, center to center, inches	$8\frac{1}{2}$
Crank pins, inches	$2 \times 2\frac{1}{8}$
web, inches	$\frac{3}{4} \times 2\frac{1}{2}$
shaft diameter, inches	2
angle, degrees	180
shaft bearings, inches	$2 \times 2\frac{1}{2}$
Fan, diameter, inches	60
width at rim, inches	$9\frac{1}{2}$
diameter air duct, inches	$30\frac{1}{2}$
Revolutions per minute	800

FEED AND FILTER TANKS.

In the engine room, with its back against the forward bulkhead, is the feed and filter tank. The main air pumps discharge into the starboard and port sides near the top. There are two compartments, containing three baskets each, of filtering material, through which the water from the air pumps is delivered on its way to a compartment to which the feed-pump suctions are attached. This feed tank is supplied with an overflow leading to the main feed tanks, a vapor pipe to

the atmosphere, a thermometer, a gauge glass and connections from the steam traps.

PUMPS.

The main feed pumps, of which there are two, are of the Blake simplex Admiralty type, bolted to the forward bulkhead of the engine room on the port and starboard sides, the hotwell being between them. They take their supply from the hotwell, from the feed tanks, and from the sea, and discharge into the main feed line to the boilers.

The auxiliary feed, fire and bilge pumps, of which there are two, of the Blake simplex Admiralty type, are located in the boiler rooms. The smaller one is fastened to the forward bulkhead of the forward fireroom on the starboard side, and the larger one is located between the two after boilers on the center line of the vessel. Each is connected to a suction manifold, with pipes leading from the sea, hotwell, reserve tanks and drainage system. They are also connected to a discharge manifold, from which pipes lead to the main feed, overboard, fire main and auxiliary feed, to the boilers in its own compartment.

A pump for draining the condenser while the main engines are idle is of the Blake simplex Admiralty type, bolted to the forward bulkhead of the engine room on the starboard side. It draws its supply from the lower part of the main air pumps and discharges into the hotwell.

A water-service pump of the same size and type as the condenser pump is located on the forward bulkhead of the engine room on the port side. It draws its supply from the sea, and delivers into the water-service pipes and also into the distiller circulating pipes.

PUMP DATA.

Main feed pumps (2).

Type	Vertical simplex.
Diameter of steam cylinder, inches	12
water cylinder, inches	8
Stroke of piston, inches	12
Gallons per double stroke	5.222
Diameter of steam pipe, inches	1¼
exhaust pipe, inches	2½
suction pipe, inches	3½
discharge pipe, inches	2½

Auxiliary Pumps (2). After Boiler Room.

Type	Vertical simplex.
Diameter of steam cylinder, inches	12
water cylinder, inches	8
Stroke of piston, inches	12
Gallons per double stroke	5.222
Diameter of steam pipe, inches	2
exhaust pipe, inches	2½
suction pipe, inches	3½
discharge pipe, inches	2½

Forward Boiler Room.

Type	Vertical simplex.
Diameter of steam cylinder, inches	10
water cylinder, inches	6
Stroke of piston, inches	12
Gallons per double stroke	2.938
Diameter of steam pipe, inches	1½
exhaust pipe, inches	1½
suction pipe, inches	3
discharge pipe, inches	2½

Condenser Pump.

Type	Vertical simplex.
Diameter of steam cylinder, inches	3½
water cylinder, inches	4
Stroke of piston, inches	4
Gallons per double stroke	.4352
Diameter of steam pipe, inch	¼
exhaust pipe, inch	¼
suction pipe, inches	2
discharge pipe, inches	1½

Sanitary Pump.

Same as condenser pump.

Distiller Circulating Pump.

Type	Horizontal duplex.
Diameter of steam cylinder, inches	4½
water cylinder, inches	2½
Stroke of piston cylinder, inches	4

Evaporator Feed Pump.

Type	Horizontal duplex.
Diameter of steam cylinder, inches	2
water cylinder, inches	1½
Stroke of pistons, inches	2½

There is an evaporator of the Davidson horizontal type, of 2,000 gallons capacity, located on port side in the after end of

the engine room, with the necessary pumps and pipe connections.

The distiller, of 800 gallons capacity, is of the Davidson vertical type, located at the side of the vessel, above the evaporator.

The air compressor is of the three-cylinder, triple compression type; it is located on the starboard side of the engine room, just aft of the engines.

The steering engine, located in the after part of the engine room, on the center line, is of the horizontal, two-cylinder type, and was built by the contractors from their own designs. It consists of a double-cylinder horizontal engine, driving a worm, which meshes with a worm wheel. The worm wheel turns on a shaft carrying the cable drum, which can be thrown in or out of connection with the worm wheel by means of a hand lever. When thrown out of gear the boat is steered by the hand wheel in the after conning tower. The controlling valve of the engine is actuated by means of a shaft leading to the pilot wheel on the bridge. There is a compensating attachment on the engine which causes it to follow the variations of the movement of the steering wheel very closely, the exact position of the tiller at all times being indicated by the telltale on the bridge.

The electric-light plant is located on the port side of the engine room, near the after bulkhead; it consists of a double-cylinder, vertical, inverted, direct-acting, piston-valve engine, directly connected to a multipolar dynamo; the whole on one base. It was furnished by the General Electric Company. This plant is of 3.6-kilowatt capacity at 800 revolutions per minute. On the port side, behind the dynamo, is the switchboard, on which is mounted the switches controlling the several circuits to different parts of the boat, and also the volt meter, ampère meter, pilot lights, fuse boxes, &c.

SPEED TRIAL.

The torpedo boat *Goldsborough* left wharf number four, Navy Yard, Puget Sound, shortly after noon, March 30, 1905, getting under way for the measured-mile course off Alkao Point at 12·12½ P. M.

The weather was clear; there was a light southerly breeze blowing, with the sea but slightly disturbed.

The run, of about 12 miles to the course, was made under forced draft, at about 20 knots speed; the upper set of range stakes was passed at 12·47 P. M.

The standardization runs consisted of three pairs of runs over the measured course. The conditions during each double run, as concerns the revolutions, were as uniform as possible, except on the last, when the revolutions dropped from 341.8 to 285.5, owing to a leak in the port feed pipe, necessitating running the evaporator and auxiliary feed pump for the forward boiler.

The time of each run over the course was taken with stop watches, and the revolutions of each engine for the elapsed time were taken from two Pratt & Whitney counters, set up on the deck aft of the engine-room hatch and actuated by means of a cam on the engine shaft below.

The results of the standardizing runs are given in the attached table.

After completing the standardizing runs, during which the speed of 27 knots, as required by the Department's order, was obtained, the hour trial was immediately run off toward Tacoma for one-half hour and return. So many air and steam leaks had developed during the progress of the trial that it was impossible to develop sufficient power to maintain more than an average of 290 revolutions, developing, according to the speed curve, a speed of 25.2 knots.

Hand-picked Cardiff coal was used on these trials and was very skilfully fired, there seldom being any black smoke showing at the funnels, and at no time was any fire observable coming from them; the steam pressure was also well maintained during the runs.

The crew carried on the trials in the engineering department was as follows, from the torpedo boat *Paul Jones:*

Engine Room.—One chief machinist, two first-class machinists, two second-class machinists, four oilers, one electrician, one water tender.

OVER MEASURED KNOT.

Number of run.	Time.		Knots.	Average revolutions per minute.	Total revolutions.		Average total revolutions.	Steam pressure at engines.	1st receiver.	2d receiver.	Vacuum.	Air pressure.	
					Stbd. engine.	Port engine.						Forward fireroom.	After fireroom.
	m.	*s.*										*inches.*	*inches.*
1st trip, East	2	50⅘	21.1	246.0	706	692	699	250	25	0	28	2	4
2d trip, West	2	52⅘	20.8	242.0	702	692	697	245	23	0	28	2	4
3d trip, East	2	26	24.7	280.2	691	673	682	255	50	15	28	3	4
4th trip, West	2	27⅕	24.5	285.7	706	694	700	250	28	12	28	3	4
5th trip, East	2	0	30.0	*341.8	704	663	683.5	260	80 s. 60 p.	30 s. 21 p.	26	4	5
6th trip, West	2	25	24.8	285.5	702	678	690	215	10	2	20	0	4

* Throttle valves wide open. Cut-off links on all runs, two-tenths down.

1·45 TO 2·45, HOUR RUN. 1·51 TO 2·40 P. M., TIME OF OBSERVATION.

Time.	Knots.	Revolutions per minute.	Total revolutions.		Average total revolutions.	Steam pressure at engines.	1st receiver.	2d receiver.	Vacuum.	Air pressure.	
			Starboard engine.	Port engine.						Forward.	Aft.
2·01	25.8	296	3,048	2,871	2,960	240	100 s. 90 p.	75 20	16	5	4
2·21	...	...	9,021	...	...	Reading on port engine not taken, as escaping steam drowned the signal.					
2·40	25.3	290.3	14,604	13,848	14,226	250	95	26	16	5	4

Forward Fire Room.—One water tender, one blower-engine tender, two firemen, two coal passers.

Aft Fire Room.—Two water tenders, two blower-engine tenders, four firemen, four coal passers. Three firemen on deck at boiler stops, three men on deck taking data.

The draught and displacement of the boat before and after the trial were as follows, pig iron being used to attain the displacement required.

	Before.	After.
Draught forward, feet and inches	4–10$\frac{1}{2}$	4– 4$\frac{1}{2}$
aft, feet and inches	9– 1$\frac{1}{2}$	9– 1
mean, feet and inches	6–11$\frac{7}{8}$	6– 8$\frac{3}{4}$
Displacement, tons	264	247
Mean displacement on trial, tons	255$\frac{1}{4}$	
Mean draught on trial, feet and inches	6–10$\frac{5}{16}$	

INSPECTION AFTER TRIAL.

Both H.P. piston-rod metallic packings leaked badly.

Port after L.P. top cylinder-head gasket gave way.

Forward pin in the port after rock arm for suspension link of L.P. valve motion snapped off close to the body of the guide block, while on the hour run.

Leak developed in feed pipe to forward boiler while on the sixth run, necessitating starting up evaporator and auxiliary feed pump.

Vapor pipe from evaporator to condenser started to leak on hour run, reducing vacuum from 28 inches to 16 inches.

Top of relief valve on head of the port after low-pressure cylinder kept lifting at intervals during the whole run, wasting a large quantity of fresh water.

All the above steam leaks, together with minor leaks, kept the engine room filled with steam, making it at times almost unendurable, but, notwithstanding this, the attendants kept at their stations and succeeded in keeping all bearings cool.

It is believed that when the boat is placed in commission, with a regular crew attached, who will naturally take interest and keep overhauling and repairing minor defects that will likely occur, the boat will prove to be a very acceptable addition to the service, provided she is not forced to extreme speed, say, not over 26 knots per hour.

NOTES ON THE PERFORMANCE OF THE THORNYCROFT BOILER IN A MONITOR.

By Lieut. W. T. Cluverius, U. S. N., Member.

To have steamed 28,000 miles since commissioning, three years ago, with nearly one-third of this time consumed in repairing a turret mount at a navy-yard dock, seems a fair test of the performance of the four Thornycroft-Daring boilers of the harbor-defence monitor *Arkansas*.

This type of water-tube boiler has been present in our torpedo craft for several years, and trouble has been found generally in the fact that the upper rows of tubes could not be drowned and corrosion in these tubes readily set in. In the later pattern of these boilers, however, these tubes are flush with the top of the steam drum and are drowned if the boiler is kept entirely full of water. This obtains on the *Arkansas*, and undoubtedly the increased life of a Thornycroft boiler tube is due to this change in design, for on the *Arkansas* not a sign of even slight corrosion has appeared until recently. In fact, no tubes had been lost, except once through careless handling of the feed when newly commissioned, until last month. Two tubes were lost at that time in the vicinity of the steaming level, the only locality in the drum where the tube ends show any evidence of corrosion.

Assuming that the life of a tube depends upon the care which it receives, and that it is not affected by the forcing of the boiler within reasonable limits, the longevity of such tube may be determined through inquiry into the attention given the boiler.

The following is the procedure on the *Arkansas*, and, whether successful or otherwise, it is at least consistent:

When the fires have been burned down and the boilers dis-

connected, the tubes are swept with a steam lance from top to bottom while there is still pressure showing in the gauge. If in port, the uptake also.

This removes much of the deposit, though not any of that which has settled in the spaces where the nests of tubes contract to enter the steam drum and the wing drum. If neglected, these spaces, for a distance of nearly a foot above the wing drums, and as much outboard from the steam drum, become entirely choked and reduce the heating surface, and therefore the efficiency, of the boiler materially.

To get at this deposit, a tin scraper, together with a stiff whisk broom, is used. Each tube is carefully gone over and the interstices cleaned out. This is necessarily a slow process, but it is time well spent.

Even this does not reach properly the inner corners where the rear nests of tubes enter the wing drums; and if the furnaces are allowed to remain any length of time without cleaning it becomes impossible to clear out these pockets without removing the rear casing of the boiler. As a remedy it is suggested that soot doors be fitted in this casing as in the front casing. Two such doors on each side, one just above the wing drum and the other well up in the casing, would meet the requirements.

As regards reaching the tubes of the middle water drum, much less difficulty has been experienced.

With this attention in port to tubes and uptakes, the boilers have steamed freely for five days without sweeping.

All of the feed water used is distilled on board, no matter whether taken from the sea or the shore. The water is given frequent nitrate tests with a standard solution at the distillers before it enters the reserve tank. The condensers receive the same test. I believe that no salt scale has ever been found in any of the four boilers.

To insure an alkaline state, a small amount of sal soda is put in each main condenser every other day while underway, and in port about a quarter pound once weekly.

To prevent the passage of grease into the boilers, no oil is

allowed in the cylinders and but little for swabbing the rods. The dynamo engines being steepled, with low-pressure cylinder below, and running in oil as they do, some oil is drawn up the low-pressure piston rod and finds its way to the feed and filter tank.

The water in this tank is constrained to pass through three thick layers of loofa—a most satisfactory filtering agent—before it reaches the feed pumps. This loofa is removed every fortnight and given a soda bath, being entirely cleansed of oil. The life of a piece of loofa is about two months. It is also used in the distiller filters with excellent results.

Once each month the boiler drums are opened, cleared of zinc baskets, baffles, etc., thoroughly wiped out and inspected. A solution of 25 pounds of sal soda is then placed in the boiler; it is closed up, filled to steaming level and fires are started. It is boiled out for six hours at a pressure of 50 pounds. The boiler is now blown down, opened and again wiped out. The drum fittings are overhauled, after which the boiler is closed and filled with water.

Judging by the present condition of the visible portions of the tubes—and it is impossible to determine the condition of a bent tube throughout its length—coupled with the uniform attention which the *Arkansas* boilers have received since installment, I would submit that the period of *complete* efficiency of a Thornycroft boiler in a vessel of the monitor type, performing the same service as the *Arkansas*, is three years. After that, watch the tubes closely.

There are two furnaces to each boiler, and the small grate area of each (24.8 square feet) necessitates a more frequent cleaning of fires than ordinarily. In free route at sea, with the average quality of coal and under natural draft, it is often found desirable to clean one furnace of each boiler or half of the whole number of furnaces every watch. With clean coal, free from clinker or under forced draft, one quarter of all the furnaces suffices.

The use of the blowers, so often demanded while steaming in squadron, through foul condition of hull or with poor fuel,

has proved very trying to the furnace linings. Not only do the walls require constant patching, but the casings themselves suffer eventually, and the least warping of the front casings produces leaky furnace-door frames and also endangers the front-lining brickwork.

In common with all boilers of this type, the proportion of heating to grate surface is large (47.3 to 1), and the gases well baffled. The *Arkansas* can steam, under normal conditions of hull and sea, 2,700 miles at a ten-knots speed.

Probably the most interesting comparison in the operation of these boilers is that of their performance with a clean hull and with a foul one, other conditions being equal.

A, is a sixty-hour trial during a run to the West Indies, clean hull, natural draft.

B, is a sixty-one-hour trial on the run North, very foul hull, natural draft.

	A.	B.
Distance run, knots	621.6	597.2
Average speed, knots	10.36	9.8
revolutions	150.1	156.4
Revolutions per knot	14.5	16.
Coal per day, this rate, tons	31.	41.
Knots per ton of coal	8.	5.7
Steaming radius, miles	2,672.	1,903.8

This shows a 25 per cent. increase in the daily coal consumption due to foulness alone; so that this factor, together with a heavy head sea, most of which is shipped, becomes a serious question in a monitor.

Following is a table which may also be of interest:

——	A.	B.	C.	D.
Revolutions per minute	206.5	197.1	180.1	186.7
Speed per hour, knots	13.5	12.7	10.7	11.6
Total horsepower developed	*2,500	1,826.6	2,080.4	2,218.9
Pounds of coal per hour	7,200	6,720	5,471	6,859.5
sq. ft. grate surface	36.4	33.9	27.5	34.6
I.H.P. per hour	2.9	3.6	2.6	3.1

*Approximated.

A, is the best hour's run the ship has made, hull clean.
B, is the two-hours official acceptance trial, hull clean.
C, a three-hours monthly speed trial, hull foul.
D, a four-hours monthly speed trial, hull foul.
All under nearly similar conditions of forced draft.

Since the three harbor-defence monitors in the North Atlantic Fleet have spent but little time defending harbors in these days of peace and are cruising in squadron with no reserve speed left them, the items of clean hulls and clean boilers, good coal and good firing, are very essential items indeed.

WEIGHTS OF MACHINERY.

By W. F. Sicard, Associate.

The subject of machinery weights is one that should be considered carefully by all shipbuilders, particularly in connection with naval work, where an excess over the contract allowance involves a penalty. It is safe to say that nearly all the vessels of the new Navy have averaged somewhat over the contract weight and have been penalized in consequence. In isolated cases it might be somewhat difficult to tell whether such excess weights were unavoidable or not, but when the weights of a considerable number of vessels of the same class are available for comparison a reasonable conclusion can be reached as to what the weights should have been for the average ship of such class. The Navy has now enough vessels completed to make such comparisons of value, and it is to the interest of both the Government and the shipbuilders if it can be made convincingly clear that such overweights as have resulted could have been avoided without imperiling the strength or durability of the machinery and without restricting, within too narrow limits, that natural variety in design which is the stepping stone of engineering progress.

A careful consideration of the returned weights of naval vessels convinces the writer that most of the instances in which the machinery installations have been overweight, up to this time, could have been avoided, and it is hoped that the following pages will make clear, in detail, where weight might have been saved.

There is, perhaps, no subject connected with the design of marine machinery concerning which more distrust and uncertainty is felt than the preliminary estimate of weights. That this should be so with vessels of unusual or untried types is not surprising; but when, as is usually the case, the

machinery is of an established type and one that has been installed in a number of vessels, the weights of which have been accurately recorded, there is no reason why, for a new vessel of similar design, a careful estimate should not approximate very closely to the finished weight.

Early in the course of design of every vessel there must be a definite and accurate assignment of weights to each of the important elements, such as armor, armament, machinery, equipment, hull, etc. Every design is a compromise in which these elements, many of them conflicting and all seeming to demand an undue share of the available weight, have been critically compared and harmonized until, when the design is finally settled, they bear a definite relation to each other, this relation being dependent upon the type of vessel and upon the service she will be required to perform. In the relation of the final design each element has its definite allotment of weight that cannot be changed except at the expense of some other element, for any variation in the weight of one element affects both the mutual relation of the others and their total weight, and the success of this part of a design depends largely upon an accurate preliminary approximation to what the finished weights will be.

In nearly all vessels of war the most important elements or characteristics are those of offence and defence, and on a given displacement all weight added to any of the other elements above their proper apportionment of the remaining available weight must be gained by reducing either or both of these vital qualities, therefore reducing the military efficiency of the vessel, and consequently, more or less, her reason for existence.

A consideration of the whole subject of the weight of all the different elements of a design and their mutual relation is much too broad to be discussed, in all its varying phases, within the limits of this paper, and therefore only what is commonly included in the term machinery weights will be considered here.

This question of machinery weight is considered so import-

ant by the Navy Department that a penalty is required for every ton in excess of the weight fixed by the contract.

The clause referring to this subject on the latest form of contract for large naval vessels of the first class reads as follows:

"* * * and that the total weight of said machinery, including engines, boilers and appurtenances, all fixtures in engine and firerooms, smoke pipes, distilling apparatus, spare parts, heating apparatus, tools in workshop, and water in boilers, condensers, pumps, pipes and stern tubes, that is to say, all weights under cognizance of the Bureau of Steam Engineering, except supplies furnished by the Government (but not including the reserve feed water in double bottom or tanks or spare parts of the machinery not usually carried on board, as enumerated in the specifications), shall not exceed * * * tons, this weight to be determined from the certified records of the actual weight of the parts of the machinery as they are sent on board the vessel to be connected up, except the weight of the contained water, which shall be calculated from the actual volumes in steaming condition, as shown on the certified drawings of the completed machinery; the weight to be calculated for salt water, except for those parts where fresh water only is used; that if said total weight be exceeded, a deduction of five hundred dollars ($500) a ton shall be made from the contract price of the vessel for each ton of weight in excess of that stipulated, and that if said total weight be exceeded by five (5) per cent., a further deduction of ten thousand dollars ($10,000) shall be made from the price of the vessel * * *."

In order to fulfill this requirement, after a contract is entered into and the work is begun every object under the cognizance of the Bureau of Steam Engineering is weighed by the contractor in the presence of a naval inspector of machinery, or one of his assistants, before being placed on board the vessel. The weights are carefully recorded by the inspector on special blank forms supplied for the purpose, and when the vessel is finished these forms are forwarded to the Bureau,

where the weights and centers of gravity are computed, and, in case there is over weight, the amount of penalty is determined in accordance with the stipulations of the contract as quoted above.

These forms, called the finished-weight sheets, are bound and kept on file at the Bureau for reference and for use in basing estimates of weights and centers of gravity of new machinery of a similar type.

Of course, all the separate items that appear on the weight sheets for naval vessels are arranged according to the Bureau's classification of weights. It often happens, however, say in the case of preliminary or unofficial estimates for new work from shipbuilders, or in statements of weight of merchant work, that the weights are given according to quite different forms of grouping and classification.

As nearly every shipbuilding firm has its own system of keeping weight returns, and as these systems vary somewhat in respect to the division of different items into groups, as well as in regard to the different subjects included under machinery weights, it is often very difficult to form comparisons, or a clear idea as to exactly what is included, when the weights of machinery installations are given by different firms in their own classification.

It would be a great advantage all around, and a step in the direction of uniformity, if outside firms would adopt the Bureau's classification of weights for all marine work. It may be urged that this cannot be conveniently done, owing to the fact that the division of work employed by the Navy Department under its bureau system does not obtain in private yards. This objection, however, can easily be overcome by adding new groups, containing the items where the jurisdiction differs, this being simpler, because the shipbuilding firms almost invariably include in their machinery weights items which, in Department work, are divided among several bureaus. Therefore, to make the change without affecting their own practice, it would only be necessary to make a rearrangement of their existing forms of classification to the extent of taking out of

their machinery weights subjects not included by the Bureau of Steam Engineering, and forming these subjects into certain new groups which, when used in naval work, could be readily credited to the bureaus concerned, and when used in their own work could be placed where desired.

Aside from the inconvenience of having different systems, it certainly involves more work and expense if, after the objects are weighed and classified in accordance with the bureau system, they must be reclassified according to another, which, by the way, is usually not as good nor as well adapted for estimate of weight or cost.

By having a definite system of weight classification for *all* machinery weights it will be found that accurate estimates and comparisons can be much more quickly made and actual weights will be much more available for use.

The system of classifying and keeping weights adopted by the Bureau of Steam Engineering has been in use for some years and has proved accurate and complete. Briefly it is as follows:

All objects under the Bureau's cognizance are divided into a number of subjects. Similar subjects or those closely related are collected into groups, the weight of each group, for each division of the ship, being kept on separate weight sheets. For the purpose of placing these groups in their proper divisions or compartments the weights are divided into—

Bow weights, including all weights of machinery forward of the forward bulkhead of main machinery compartments.

Stern weights, including all weights abaft the after bulkhead of main machinery compartments.

The weights between the forward and after bulkheads of main machinery compartments are divided according to the actual transverse bulkheads in the ship and named according to location, "*After*," "*Middle*," or "*Forward*," "*Fireroom Weights;*" "*Forward*" or "*After*" "*Engine-room Weights.*" Should there be bunkers or auxiliary machinery rooms in this space (separated from it by transverse bulkheads) an appropriate title for the space so included is assumed.

By "main machinery compartments" is meant all space included between the forward and after bulkheads inclosing the main engines and boilers. All bulkheads are supposed to extend upward indefinitely and to each side of the ship.

The individual subject division of all the machinery weights and the arrangement of the same into groups, according to the Bureau's classification, is as follows:

GROUP I.—MAIN ENGINE CYLINDERS.

Subject.

1. H.P. cylinders, including cylinder casings, liners, covers; valve chests, liners and covers; bolts and nuts.
2. 1st I.P. cylinders, including same.
3. 2d I.P. cylinders, including same.
4. L.P. cylinders, including same.
5. Relief and drain valves and receiver safety valves for main-engine cylinders.
6. Main receiver piping.
7. Main piston rod and valve-stem stuffing boxes and packing.

GROUP II.—SHAFTING.

Subject.

1. Main crank shafts.
2. Main line shafts.
3. Main propeller and stern-tube shafts, with nut and keys.
4. Main thrust shafts.
5. Main shaft couplings (special), friction wheels, brakes and clutch couplings.
6. Main shaft coupling bolts and nuts.

GROUP III.—MAIN ENGINE FRAMING AND BEARINGS.

Subject.

1. Bedplates, including crank-shaft brasses, caps and bolts, and foundation bolts and nuts.
2. Columns and tie rods and housings, with bolts and nuts.
3. Crosshead guides, with bolts and nuts.
4. Thrust bearings, with bolts and nuts.
5. Line-shaft bearings, with bolts and nuts.
6. Stern tube, stern-bracket and strut bearings, with bolts and nuts. (Not including tube, bracket or strut.)
7. Bulkhead stuffing boxes around shafting (complete).

GROUP IV.—RECIPROCATING PARTS OF MAIN ENGINES.

Subject.

1. H.P. pistons, complete.
2. 1st I.P. pistons, complete.
3. 2d I.P. pistons, complete.
4. L.P. pistons, complete.
5. Piston rods, with nuts, keys, etc.
6. Crossheads complete, including slippers.
7. Connecting rods; rods, brasses, straps, bolts and nuts, complete.

GROUP V.—MAIN ENGINE VALVE GEAR.

Subject.

1. Eccentrics.
2. Eccentric straps and bolts.
3. Eccentric rods, brasses and bolts.
4. Links and blocks, complete.
5. H.P. valves, complete.
6. 1st I.P. valves, complete.
7. 2d I.P. valves, complete.
8. L.P. valves, complete.
9. Valve stems, brasses, nuts, keys and balance pistons.
10. Valve-stem crossheads.
11. Valve rock shafts.
12. Suspension links, crosshead guides, radius rods and other gear.
13. Reversing engines and gear, complete, with shafts, brackets and arms.

GROUP VI.—MAIN CONDENSERS.

Subject.

1. Main condensers, shells, heads, tubes, etc., complete.

GROUP VII.—MAIN AIR AND CIRCULATING PUMPS.

Subject.

1. Main air pumps, barrel, liner, covers, bucket, rods, etc., complete to piping.
2. Main air-pump engines, complete to piping.
3. Circulating pump, complete to piping.
4. Circulating-pump engines, complete to piping.

GROUP VIII.—PROPELLERS.

Subject.

1. Main propellers, without shaft nut, but with cap and all other bolts and nuts.

GROUP IX.—BOILERS.

Subject.

1. Main boilers, including shells, butt straps, rivets, heads, furnaces, combustion chambers, tubes, tube sheets, and all rivets and braces, and manhole plates, and, in the case of water-tube boilers, including Subject 6, Group XIV.
2. Auxiliary boilers, including same.

GROUP X.—BOILER FITTINGS.

(*a*) Main boilers. (*b*) Auxiliary boilers.

Subject.

1. Grate bars and bearers.
2. Bridge walls and fittings.
3. Furnace fronts, doors, ash-pit doors, etc., lazy bars.
4. Ash pans.
5. Other furnace fittings.
6. Dry pipes, and all other internal pipes, circulating apparatus hangers, etc.
7. Zincs, hangers, baskets and bolts.
8. Water gauges and cocks; safety, sentinel and air valves; drain, air, soda and other cocks attached to boilers.
9. Boiler fastenings for securing boilers in ship (not including saddles).

GROUP XI.—SMOKE PIPES AND UPTAKES.

(*a*) Main boilers. (*b*) Auxiliary boilers.

Subject.

1. Uptakes, including all weights from boiler to base of smoke pipe.
2. Smoke pipes, including all weights of pipes, guys, etc.

GROUP XII.—STEAM AND EXHAUST PIPES AND VALVES.

Subject.

1. Main steam pipes and valves, including pipe, flanges, bolts and nuts, hangers and fittings.
2. Auxiliary steam pipes and valves, including same.
3. Branch steam pipes to machinery under Bureau of Steam Engineering, including same.
4. Main exhaust piping.
5. Auxiliary exhaust piping.
6. Branches of exhaust piping to machinery under Bureau of Steam Engineering.
7. Escape pipes, whistles, mufflers, etc.

GROUP XIII.—SUCTION AND DISCHARGE PIPES AND VALVES.

Subject.

1. Main injection (bilge and sea) pipes, valves and fittings.
2. Main outboard delivery pipes, valves and fittings.
3. Main air-pump suction and discharge pipes, valves and fittings.
4. Main and auxiliary feed-pump pipes, valves and fittings.
5. Other auxiliary pump pipes, valves and fittings (except water-service pipes).
6. Pump relief valves.
7. Blowpipes and valves outside of boilers, and connections for pumping out boilers.
8. Drain pipes and traps (to machinery under Bureau of Steam Engineering).

NOTE.—Branches from pumps to fire main and main drain pipes will be taken up to flanges on those pipes. All suction pipes to bilges of machinery compartments will be included.

GROUP XIV.—LAGGING AND CLOTHING.

Subject.

1. Lagging and clothing of main cylinders.
2. Lagging and clothing of cylinders of auxiliary engines, feed heaters, etc.
3. Lagging and clothing of main condensers.
4. Lagging and clothing of auxiliary condensers.
5. Lagging and clothing of feed tanks.
6. Lagging and clothing of main boilers.
7. Lagging and clothing of auxiliary boilers.
8. Lagging and clothing of pipes and valves.
9. Lagging and clothing of dynamo condensers and pumps.

GROUP XV.—FLOORING, GRATINGS, ETC.

Subject.

1. Floor plates, supports, bolts, etc.
2. Gratings, platforms, supports, bolts, etc.
3. Ladders, supports, bolts, etc.
4. Hand rails, supports, bolts, etc.
5. Guards, supports, bolts, etc.

GROUP XVI.—AUXILIARIES.

Subject.

1. Feed pumps for main boilers.
2. Feed pumps for auxiliary boilers.
3. Fire and bilge pumps.
4. Water-service pumps.
5. Other auxiliary steam pumps (not including evaporator pumps).
6. Auxiliary condenser with air and circulating pumps.
7. Shaft pumps.
8. Turning engines and gear.
9. Ash hoists.
10. Blowers and engines (not including ventilating blowers).
11. Coal winches.
12. Other auxiliary engines under Bureau of Steam Engineering.
13. Dynamo condensers, with air and circulating pumps.

GROUP XVII.—FITTINGS AND GEAR.

Subject.

1. Feed and filter tanks, fittings, and vapor pipes.
2. Oil, tallow, soda and waste tanks and fittings.
3. Water-service pipes, main and auxiliary machinery.
4. Lubricating gear, main and auxiliary machinery.
5. Lifting gear for handling parts of machinery.
6. Gear for working valves from decks.
7. Gauges, thermometers, clocks, counters, telegraphs, etc.
8. Separators.
9. Gear for operating engines.
10. Counter gear.
11. Indicator gear.
12. Governors.
13. Air ducts (not including ventilating ducts).
14. Feed-water heaters, grease extractors, and other fittings.
15. Other gear.

GROUP XVIII.—WATER.

Subject.

1. Water in condensers. (Taken full on salt-water side only.)
2. Water in feed tanks. (Taken two-thirds full.)
3. Water in stern tubes.
4. Water in main boilers.
5. Water in auxiliary boilers.

{ Taken 4 inches above highest heating service for Scotch boilers and at working level of other boilers.

6. Water in all auxiliary pumps, pipes and auxiliary machinery necessarily in use when running engines at full power.

GROUP XIX.—STORES, TOOLS AND SPARE PARTS.

Subject.

1. Tools in workshop, including engine or motor.
2. Other tools, with racks, etc.
3. Spare parts, with bolts, etc., for securing.
4. Stores supplied by contractors.
5. Hose and racks.
6. Indicators and instruments not always in use.

GROUP XIXa.—GOVERNMENT STORES.

Subject.

1. Stores supplied by Government, under titles B and Y.

GROUP XX.—MISCELLANEOUS MACHINERY, ETC.

Subject.

1. Distilling plant, including distillers, evaporators, pumps, pipes, lagging and clothing, fittings and water.
2. Heating plant, including radiators, pipes, valves, lagging and clothing, and fittings.
3. Hydraulic pumping plant, including pumps, accumulators and all fittings, and water.
4. Steam turret-turning plant.
5. Refrigerating plant, including ice machine and all refrigerating pipes.
6. Air-compressor plant, under Steam Engineering.

GROUP XXI.—OTHER MISCELLANEOUS MACHINERY, ETC.

This group is composed of certain subjects which are not under the Bureau of Steam Engineering, and its weights are not included.

GROUP XXII.—CONNECTIONS UNDER STEAM ENGINEERING TO "OTHER MISCELLANEOUS MACHINERY."

Subject

1. Branch steam pipes and valves.
2. Branch exhaust pipes and valves.
3. Suction and discharge pipes and valves.
4. Drain pipes, traps and valves.
5. Lagging and clothing.
6. Water.
7. Lubricating gear and water service.

At first glance this system of classification may appear to be in unnecessary detail, but, while seemingly elaborate, it is in reality very simple and flexible, permitting the finished weights to be so recorded that the actual weight of even a small object or the part of a subject can be quickly found if desired, thus being of use in estimating for cost of new parts. When used for estimate it offers several different methods for computing the weights. It permits either a rough and general estimate to be quickly made, based on the total weight, or an estimate based on per cent. of compartment weights in similar vessels, or a group estimate, or a subject estimate, this latter being capable of great elaboration and refinement in detail.

It is often useful to know the machinery weight in any compartment or its per cent. of the total weight, and if this division of weight is made at the time of estimating, it really involves very little extra work, besides permitting a much closer estimate to be made.

It is the writer's opinion that there are no formulae of any use for estimating machinery weights. A preliminary estimate of the total weight necessary for the machinery installation of a new vessel is usually required before the size of various objects and their location is definitely determined. This preliminary estimate must often be made when possibly only the horsepower, the general type of engines and boilers and a few very indefinite general particulars are known, and there is usually at this stage of the work no drawing for locat-

ing the machinery and getting the length of shafting and such items that very much affect the weight.

Under such conditions it is manifestly impossible to make any kind of a detailed estimate, and the best way to figure this first and preliminary weight is to base it on pounds per indicated horsepower, selecting a constant from a table of weights of machinery having similar salient features. This gives a more accurate result than might be expected for what might be minor inaccuracies in group and subject were a detailed estimate made, tend to average, and, in dealing with the larger sum, they disappear and are lost. Later, when the type and size of the machinery is more definitely settled, and the drawings are far enough advanced to scale for size of parts and location of centers of gravity, another estimate should always be made. This time it is well to estimate on the subjects in detail, according to the classification, bearing in mind that the general tendency is always to underestimate, and checking the new weights by subject, group and compartment comparison with other similar vessels, and checking their total with the first estimate. This detailed estimate, with its cross checks and the facility it offers for comparison, should show at once where, if at all, an error has been made, and should show to what particular group, or item, or subject, an excess or deficiency is due.

This estimate, based on individual subjects, does not take as long to make as might be supposed. A glance at the accompanying finished-weight sheet, which shows group IV and all its subjects, will make it apparent how closely the weight of a new engine of a similar design could be figured from such detail as this sheet shows.

The method of keeping weights adopted by the Bureau of Steam Engineering is as follows:

The actual weight of each subject in detail, and the distances of its center of gravity from fixed vertical and horizontal planes, are entered on the following blank forms, called the finished-weight sheets.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Subject.	For estimate, weight per cubic inch: Cast iron = 0.261 lbs. Wrought iron = 0.28 lbs. Cast steel = 0.283 lbs. Wrought steel = 0.283 lbs. Composition = 0.316 lbs. Water, to suit temperature. Description of part.	Weight in pounds.	Center of gravity (aft = +, forward = —). Horizontal from frame 60.	Vertical above base line.
1.	H.P. piston, complete:			
	S. Piston, 9 inches diameter	46		
	4 bolts and nuts, $\frac{3}{4} \times 2\frac{1}{2}$ inches	3		
	Piston ring	2.5		
	P. Piston, details as above	52.5		
1.	H.P. pistons, total	104	+ 17.38	11.34
2.	I.P. piston, complete:			
	S. Piston, $15\frac{1}{2}$ inches diameter	86.5		
	6 bolts and nuts, $\frac{3}{4} \times 2\frac{1}{2}$ inches	4.5		
	Piston ring	5		
	P. piston, details as above	96.5		
2.	I.P. pistons, total	192.5	+ 20.29	11.32
4.	L.P. piston, complete:			
	S. Piston, $25\frac{1}{2}$ inches diameter	190		
	8 bolts and nuts, $\frac{3}{4} \times 2\frac{1}{2}$ inches	6		
	Piston ring	10		
	P. piston, details as above	206		
4.	L.P. pistons, total	412	+ 22.70	12.63
5.	Piston rods with nuts and keys:			
	S. 3 main piston rods, 3 ft. $6\frac{5}{16}$ in. over all	187.5		
	3 nuts, $1\frac{5}{8}$ inches diameter	8		
	P. 3 main piston rods and nuts	195.5		
5.	Piston rods with nuts and keys, total	391	+ 19.94	9.76
6.	Crossheads and slippers, complete:			
	S. 3 crosshead brasses, $3\frac{3}{8}$ in. dia. $\times \frac{5}{8}$ in	87		
	6 bolts and nuts, $1\frac{3}{8} \times 9\frac{9}{16}$ inches	33		
	3 crosshead slippers, $8 \times 8\frac{1}{2}$ inches surface	96		
	12 tap bolts, $\frac{3}{4} \times 1\frac{1}{2}$ inches	4.5		
	P. 3 crosshead brasses and slippers, details as above	220.5		
6.	Crossheads and slippers, total	441	+ 19.86	9.06
7.	Connecting rods, complete:			
	S. 3 connecting rods, 4 ft. $4\frac{1}{2}$ in. bet. centers	373		
	3 connecting-rod brasses	294		
	6 bolts and nuts, $1\frac{1}{4} \times 14$ inches	54		
	P. 3 connecting rods, details as above	721		
7.	Connecting rods, brasses, etc., total	1,442	+ 19.62	7.48

When the vessel is completed, the finished-weight sheets are forwarded to the Bureau, where the weights in detail, with subjects properly collected into groups, are entered on the

following blank form, and the moments computed for each subject of each group, from which are obtained the centers of gravity of each group.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Group.	Description of part.	Total weights.		Horizontal.		Vertical.	
		Pounds.	Tons.	From frame 78.5.	Moments.	Above base line.	Moments.
III.	Bedplates	143,621	64.12	28.5	1,827.42	7.5	480.9
	Columns and tie rods	76,329	34.08	27.8	947.42	13.7	466.9
	Crosshead guides	4,779.5	2.13	21.6	46.01	16.1	34.29
	Crosshead guides	4,684.5	2.09	21.6	54.14	16.1	33.65
	Crosshead guides	4,614.5	2.06	32.9	67.77	16.2	33.37
	Crosshead guides	4,834.5	2 16	32.9	71.06	16.2	34.99
	Crosshead guides	3,619.5	1.61	15.9	25.60	16	25.76
	Crosshead guides	3,627 5	1.62	15.9	25.76	16	25.92
	Crosshead guides	3,544 5	1 58	39.6	62.57	16.3	25.75
	Crosshead guides	3,547.5	1.58	39 6	62.57	16.3	25.75
	Thrust bearings	35,313.2	15 77	52	820 04	8	126.16
	Total	288,516	128.8	31.14	4,010.36	10.2	1,313.44
IV.	H.P. piston	3,628	1.62	21.5	34.83	20.5	40.5
	I.P. piston	5,534.5	2.47	32.8	81.02	22.9	56.56
	L.P. piston	3,637	1.62	15.8	25.60	20.8	33.7
	L.P. piston	3,577	1.6	15.8	25.28	20 8	33.28
	L.P. piston	3,647	1.63	39.5	64.39	22.9	37.33
	L.P. piston	3,637	1.62	39.5	63.99	22 9	37.1
	H.P. piston rod	1,634	.73	21.6	15.77	21.8	15.91
	Piston rods	1,639	.73	21.6	15.77	21.8	15.91
	Piston rods	1,606	.72	32.9	23.69	19.6	14.11
	Piston rods	1,606	.72	32 9	23.69	19.6	14.11
	Piston rods	1,557	.69	15.9	10 97	17.6	12.14
	Piston rods	1,517	.68	15.9	10.81	17.6	11.97
	Piston rods	1,532	.68	39.6	26.93	19 6	13.33
	Piston rods	1,547	.69	39.6	27.32	19.6	13.52
	Crossheads	2,436.5	1.09	21.6	23.54	17.9	19.51
	Crossheads	2,440 5	1 09	21.6	23.54	17.9	19.51
	Crossheads	2,438.5	1.09	32.9	35.86	15.8	17.22
	Crossheads	2,447.5	1.09	32.9	35.86	15.8	17.22
	Crossheads	2,237 5	1	15.9	15.9	13.8	13.8
	Crossheads	2,238 5	1	15.9	15.9	13.8	13.8
	Crossheads	2,245.5	1	39.6	39.6	15.9	15.9
	Crossheads	2,238.5	1	39.6	39.6	15.9	15.9
	Connecting rod	7,234	3.23	21.7	70.09	13	41.99
	Connecting rod	7,209	3 22	21.7	69.87	13	41.86
	Connecting rod	7,214	3.22	33	106.26	11	35.42
	Connecting rod	7,189	3.21	33	105.93	11	35.31
	Connecting rod	7,209	3.22	16	51.52	8.9	28.66
	Connecting rod	7,139	3 19	16	51.04	8.9	28.39
	Connecting rod	7,179	3.20	39.7	127.04	11.1	35.52
	Connecting rod	7,144	3.19	39 7	126.64	11.1	35.41
	Total	112,538.5	50.24	27.63	1,388.25	15.22	764.89
V.	Eccentrics	9,979	4.46	27.6	123.1	8.3	37.02
	Eccentric straps	13,493	6.02	27.6	166.15	8.4	50.57
	Eccentric rod	382	.17	25.7	4.37	13.8	2.35
	Eccentric rod	384	.17	25.7	4.37	13.8	2.35
	Eccentric rod	402	.18	26.1	4.7	13.8	2.48
	Eccentric rod	405	.18	26.1	4 7	13.8	2.48
	Eccentric rod	468	.21	28.1	5.9	14.3	3
	Eccentric rod	471	.21	28.1	5.9	14.3	3
	Eccentric rod	469	.21	10.7	2.25	14	2.94
	Eccentric rod	469	.21	10.7	2.25	14	2.94
	Eccentric rod	469	.21	44.8	9.41	13.7	2.88
	Eccentric rod	470	.21	44.8	9.41	13.7	2.88
	Eccentric rod	494	.22	28.5	6.27	13.4	2.95
	Eccentric rod	496	.22	28.5	6.27	13.4	2.95
	Eccentric rod	500	.22	11.1	2.44	13.9	3.06
	Eccentric rod	496	.22	11.1	2.44	13.9	3.06
	Eccentric rod	495	.22	44.4	9.77	14.1	3.1
	Eccentric rod	497	.22	44.4	9.77	14.1	3.1
	Links and blocks	427	.19	25.7	4.88	16.5	3 14

All the groups are then combined on sheets similar to the following, and the centers of gravity computed for the sums of all the groups in each compartment.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Group.	Compartment.	Total weight, in tons.	Horizontal moments: foot-tons. Forward of frame No. 78½.	Vertical moments: foot-tons. Above base line.
	Forward boiler room.			
IX.	Boilers	176.02	22,297.99	2,411.48
X.	Boiler fittings	64.43	8,153.44	628.68
XI.	Smoke pipes and uptakes	46.27	6,118.56	1,829.7
XII.	Steam and exhaust pipes and valves	14.76	1,874.71	358.24
XIII.	Suction and discharge pipes and valves.	13.57	1,683.59	142.32
XIV.	Lagging and clothing	1.79	227.49	35.
XV.	Flooring, gratings, etc	11.27	1,464.07	82.63
XVI.	Auxiliaries	11.82	1,496.68	277.8
XVII.	Fittings and gear	1.45	186.14	32.7
XIX.	Stores, tools, spares	13.49	1,826.02	169.54
XX.	Miscellaneous machinery	2.68	347.06	91.3
XXII.	Pipes connecting to XXI	.94	142.86	21.25
	Total without water	358.49	45,818.61	6,080.64
XVIII.	Water	42.59	5,405.95	594.92
	Total with water	401.08	51,224.56	6,675.56
	Middle boiler room.			
IX.	Boilers	117.50	9,987.11	1,609.75
X.	Boiler fittings	42.95	3,650.87	419.06
XI.	Smoke pipes and uptakes	69.1	5,879.45	3,163.3
XII.	Steam and exhaust pipes and valves	12.	1,019.55	276.82
XIII.	Suction and discharge pipes and valves.	10.89	929.50	121.04
XIV.	Lagging and clothing	1.23	104.26	24.46
XV.	Flooring, gratings, etc	7.41	629.65	56.07
XVI.	Auxiliaries	9.01	760.4	185.6
XVII.	Fittings and gear	.96	81.39	22.48
XIX.	Stores, tools, spares	6.43	850.12	136.72
XX.	Miscellaneous machinery	1.48	122.68	51.26
	Total without water	278.96	24,014.98	6,066.56
XVIII.	Water	28.49	2,419.87	398.44
	Total with water	307.45	26,434.85	6,465.

Finally, the weights and moments of all the compartments are collected on a "summary" sheet, and the center of gravity of the entire machinery installation computed.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Compartment.	Total weight.	Horizontal moments: foot-tons.			Vertical moments: foot-tons.	
		Forward of frame No. 78½.		Abaft frame No. 78½.	Above base line.	
	Tons	*Distances*	*Moments*	*Moments*	*Distances*	*Moments*
Summary without water:						
Stern	139.1	...	...	16,650.72	...	1,463.5
Both engine rooms	677.38	...	...	18,128.65	...	10,832.62
Bunker	50.92	...	465.23	...	...	1,018.26
After boiler room	359.76	...	15,663.11	...	...	6,149 06
Middle boiler room	278.96	...	24,014.98	...	...	6,066.56
Forward boiler room	358.49	...	45,818.61	...	...	6,080.64
Bow	33.44	...	5,982.74	...	...	821.73
	...	...	91,944.67	...	...	...
	...	...	34,779.37	34,779.37	...	...
Total without water	1,898.05	30 12	57,165.30	...	17.09	32,432.37
Summary with water:						
Stern	141.25	...	...	16,869.38	...	1,482.42
Both engine rooms	733.50	...	...	19,599.87	...	11,551.82
Bunker	51.21	...	467.55	...	...	1,022.04
After boiler room	409.73	...	17,507.35	...	...	6,750.03
Middle boiler room	307.45	...	26,434.85	...	...	6,465.
Forward boiler room	401.08	...	51,224.56	...	...	6,675.56
Bow	34.08	...	6,082.99	...	...	826.47
	...	...	101,717.3	...	...	...
	...	...	36,469.25	36,469.25	...	...
Total with water	2,071.3	31.5	65,248.05	...	16.79	34,773.34
XIXa.—Government stores	40.	...	...	...	...	...
Total machinery weight	2,111.3	...	...	...	...	...

We have now been working gradually upward, first with subjects, then groups, then compartments, until the total for the ship is reached. As the centers of gravity are rarely used, in comparison with the weights, the following sheet has been evolved, and it is the most useful and convenient method of stating the weights, showing, as it does completely, both the weights of all the groups and their relative location in the vessel.

For the purpose of comparing the weights of marine machinery installation, a number of cases have been selected in which different types of vessels having machinery of different powers are employed. In all cases the vessels compared are sister ships, and those built by different firms are selected, so as to show in what way and to what extent the machinery weights are affected by the differences that exist in engineering practice. In all of the following cases, except the three ships of the *Maine* class, the builders have developed their

ACTUAL MACHINERY WEIGHTS OF U. S. S. *****.

Groups.	Stern.	After engine room.	After bunker.	After fireroom.	After middle fireroom.	Forward fire-room.	Bow.	Not placed.	Total.
I.—Main-engine cylinders	...	175.03	...	...	...	...	...	...	175.03
II.—Shafting	78.54	44.16	...	...	...	...	...	...	122.7
III.—Main-engine framing and bearings	6.43	128.8	...	...	...	...	...	...	135.23
IV.—Reciprocating parts of main engines	...	50.24	...	...	...	...	...	...	50.24
V.—Main-engine valve gear	...	38.2	...	...	...	...	...	...	38.2
VI.—Main condensers	...	47.55	...	...	...	...	...	...	47.55
VII.—Main air and circulating pumps	...	31.29	...	...	...	...	...	...	31.29
VIII.—Propellers	28.23	...	...	...	...	...	...	...	28.23
IX.—Boilers	...	...	...	176.08	117.5	176.02	...	...	469.6
X.—Boiler fittings	...	...	...	64.44	42.95	64.43	...	...	171.82
XI.—Smoke pipes and uptakes	...	...	...	47.44	69.1	46.27	...	...	162.81
XII.—Steam and exhaust pipes and valves	...	23.41	4.04	18.95	12.	14.77	1.05	...	74.22
XIII.—Suction and discharge pipes and valves	.15	36.3	1.47	15.27	10.89	13.57	.97	...	78.62
XIV.—Lagging and clothing	...	9.11	.38	1.99	1.24	1.79	.31	...	14.82
XV.—Flooring, gratings, etc	1.81	17.56	.92	11.86	7.41	11.26	...	...	50.82
XVI.—Auxiliaries	...	25.97	...	11.81	9.01	11.82	4.69	...	63.3
XVII.—Fittings and gear	.31	35.66	.21	1.45	.97	1.44	1.21	...	41.25
XIX.—Stores, tools, spares	13.86	8.21	16.15	7.92	6.42	13.49	.36	...	66.41
XX.—Miscellaneous machinery	7.16	5.37	27.75	2.54	1.48	2.68	16.31	...	63.29
XXI.—Other miscellaneous machinery	...	...	...	...	...	...	...	...	...
XXII.—Pipes connecting to XXI	2.62	.53	...	...	...	.94	8.53	...	12.62
Total without water	139.11	677.39	50.92	359.75	278.97	358.48	33.43	...	1,898.05
XVIII.—Water	2.15	56.13	.29	42.97	28.49	42.59	.64	...	173.26
Total with water	141.26	733.52	51.21	402.72	307.46	401.07	34.07	...	2,071.31
XIXa.—Government stores	...	...	...	...	...	...	...	40	40.
Total	...	...	...	...	...	...	...	...	2,111.

designs and plans from the same plans and specifications of the Department; where, therefore, there are deviations from these plans and differences between their own designs, these changes and differences may fairly be assumed to be the result of definite and well-considered intention.

The general characteristics of the machinery selected for comparison are as follows: In every case twin screw. Where power is given it is the combined power of both main engines. Where weights are given they are in every case the weights of all objects of that particular kind.

We will take the following vessels:

Four monitors of about 2,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Arkansas*, *Nevada*, *Florida*, *Wyoming*.

Four cruisers of about 4,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Denver*, *Des Moines*, *Tacoma*, *Cleveland*.

Three battleships of about 10,000 I.H.P., 180 pounds boiler pressure, Scotch boilers. *Illinois*, *Alabama*, *Wisconsin*.

Three battleships of about 16,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Maine*, *Missouri*, *Ohio*.

Two armored cruisers of about 23,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Pennsylvania*, *Maryland*.

Three destroyers of about 8,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Chauncey*, *Paul Jones*, *Worden*.

Before proceeding to a detailed consideration of the machinery weights by groups, let us compare the weights of the main engines alone. Under the term main engines, are included—

Group I.—Main engine cylinders entire group.

Group II.—Shafting; crank shaft.

Group III.—Framing and bearings; bedplates, columns, tie rods and crosshead guides.

Group IV.—Reciprocating parts; entire group.

Group V.—Valve gear; entire group.

The actual weights of these parts are as follows:

XU

The following tables give, in their proper groups, the actual weights of all objects, under the cogni

Table I.—

Groups.	Monitors. Engines, 3-cylinder triple. Water-tube boilers.				Sma Engines, 4-cylinde	
	Thornycroft. — *Arkansas.*	Niclausse. — *Nevada.*	Mosher. — *Florida.*	B. & W. — *Wyoming.*	B. & W. — *Denver.*	B. & W — *DesMoi*
I.—Main-engine cylinders	16.56	17.013	14.888	16.717	28.027	26.76
II.—Shafting	8.66	7.912	8.655	8.875	20.91	20.04
III.—Main-engine framing and bearings	12.2	11.328	11.506	12.133	25.059	25.18
IV.—Reciprocating parts of main engines	3.51	3.729	3.488	3.836	7.604	7.29
V.—Main-engine valve gear	3.72	4.262	3.987	5.045	7.489	7.03
VI.—Main condensers	12.11	13.347	12.347	13.882	12.302	12.00
VII.—Main air and circulating pumps	3.44	3.005	2.807	3.285	5.262	5.92
VIII.—Propellers	3	2.854	3.658	3.351	8.146	5.27
IX.—Boilers	59.65	55.381	35.492	55.497	95.757	96.35
X.—Boiler fittings	16.3	42.953	14.579	27.461	38.473	38.18
XI.—Smoke pipes and uptakes	13.87	20.324	17.002	11.423	26.59	26.88
XII.—Steam and exhaust pipes and valves	12.745	12.34	12.994	9.694	14.978	15 98
XIII.—Suction and disch. pipes and valves	13.767	12.817	13.01	11.72	18.204	25.33
XIV.—Lagging and clothing	2.673	2.387	3.102	4.15	4.414	6.12
XV.—Flooring, gratings, etc	7.823	8.536	6.399	12.78	14.441	16.04
XVI.—Auxiliaries	9.34	9.018	9.481	9.562	20.903	20.35
XVII.—Fittings and gear	10.985	9.085	8.783	9.976	16.203	14.67
XIX.—Stores, tools, spares	7.834	8.35	6.504	12.39	17.249	17.86
XX.—Miscellaneous machinery	13.571	10.358	13.577	11.315	21.449	25.71
XXII.—Pipes connnct'g to other misc. mach.	6.418	4.456	5.808	4.472	3.862	1.62
Total without water	238.176	259.455	208.067	247.564	407.322	414.71
XVIII.—Water	13.312	17.029	14.003	17.612	37.492	37.40
Total with water	251.488	276.484	222.07	265.176	444.814	452.11

er the cognizance of the Bureau of Steam Engineering, for the vessels selected. A comparis

Table I.—TOTAL WEIGHTS OF MACHINERY. (ALL WEIGHTS ARE IN TONS.)

	SMALL CRUISERS.			BATTLESHIPS.				
	gines, 4-cylinder triple. Water-tube boilers.			Engines, 3-cylinder triple.				
V.	B. & W.	B. & W.	B. & W.	Scotch.	Scotch.	Scotch.	Niclausse.	T
r.	*DesMoines.*	*Tacoma.*	*Cleveland.*	*Illinois.*	*Alabama.*	*Wisconsin.*	*Maine.*	
7	26.762	30.625	28.023	91.03	87.16	96.01	107.727	
	20.047	20.475	20.145	59.99	59.27	60.61	83.214	
9	25.188	26.925	25.085	73.22	73.85	82.18	102.477	
4	7.29	7.796	7.404	26.58	24.5	27.06	34.268	
9	7.034	7.521	7.259	24.52	24.57	27.758	24.677	
2	12.007	12.454	11.938	23.69	22.96	24.28	31.46	
2	5.928	4.95	5.582	15.95	16.18	13.05	20.734	
6	5.279	4.565	5.462	23.64	19.5	18.323	20.922	
7	96.35	96.202	96.169	331.76	335.94	327	476.29	
3	38.184	37.492	36.391	60.86	60.57	55.34	74.455	
	26.882	27.01	30.848	69.49	60.57	59.34	148.12	
8	15 988	13.941	15.539	40.06	30.827	41.702	67.361	
4	25.336	20.837	29.052	32.57	31.63	38.164	52.143	
4	6.123	4.468	4.996	20.87	28.18	31.235	16.58	
1	16.044	17.563	18.699	35.32	34.62	35.302	34.348	
3	20.353	18.284	18.014	35.39	31.94	35.229	56.594	
3	14.679	16.789	15.665	27.42	27.953	35.593	39.689	
9	17.869	15.454	16.764	29.97	38 94	40.191	41.706	
9	25.711	19.614	23.199	42.92	37.637	39.853	48.784	
2	1.626	2.224	3.109	16.84	10.393	19.559	18.433	
2	414.71	405.189	419.343	1,082.09	1,057.18	1,107.879	1,490.976	
2	37.401	36.737	38.144	196.58	156.26	169.787	112.037	
4	452.111	441.926	457.926	1,278.67	1,213.44	1,277.666	1,603.013	

cts, under the cognizance of the Bureau of Steam Engineering, for the vessels selected. Acomparison of the

Table I.—TOTAL WEIGHTS OF MACHINERY. (ALL WEIGHTS ARE IN TONS.)

SMALL CRUISERS. Engines, 4-cylinder triple. Water-tube boilers.				BATTLESHIPS. Engines, 3-cylinder triple.				Engines,
B. & W. — *Denver.*	B. & W. — *DesMoines.*	B. & W. — *Tacoma.*	B. & W. — *Cleveland.*	Scotch. — *Illinois.*	Scotch. — *Alabama.*	Scotch. — *Wisconsin.*	Niclausse. — *Maine.*	Thornycroft. — *Missouri.*
28.027	26.762	30.625	28.023	91.03	87.16	96.01	107.727	112.289
20.91	20.047	20.475	20.145	59.99	59.27	60.61	83.214	74.527
25.059	25.188	26.925	25.085	73.22	73.85	82.18	102.477	101.336
7.604	7.29	7.796	7.404	26.58	24.5	27.06	34.268	34.775
7.489	7.034	7.521	7.259	24.52	24.57	27.758	24.677	24.895
12.302	12.007	12.454	11.938	23.69	22.96	24.28	31.46	30.151
5.262	5.928	4.95	5.582	15.95	16.18	13.05	20.734	18.839
8.146	5.279	4.565	5.462	23.64	19.5	18.323	20.922	24.124
95.757	96.35	96.202	96.169	331.76	335.94	327	476.29	265.365
38.473	38.184	37.492	36.391	60.86	60.57	55.34	74.455	92.317
26.59	26.882	27.01	30.848	69.49	60.57	59.34	148.12	119.458
14.978	15 988	13.941	15.539	40.06	30.827	41.702	67.361	56.56
18.204	25.336	20.837	29.052	32.57	31.63	38.164	52.143	43
4.414	6.123	4.468	4.996	20.87	28.18	31.235	16.58	11.321
14.441	16.044	17.563	18.699	35.32	34.62	35.302	34.348	45.129
20.903	20.353	18.284	18.014	35.39	31.94	35.229	56.594	46.812
16.203	14.679	16.789	15.665	27.42	27.953	35.593	39.689	34.682
17.249	17.869	15.454	16.764	29.97	38 94	40.191	41.706	29.855
21.449	25.711	19.614	23.199	42.92	37.637	39.853	48.784	53.126
3.862	1.626	2.224	3.109	16.84	10.393	19.559	18.433	17.909
407.322	414.71	405.189	419.343	1,082.09	1,057.18	1,107.879	1,490.976	1,236.47
37.492	37.401	36.737	38.144	196.58	156.26	169.787	112.037	80.021
444.814	452.111	441.926	457.926	1,278.67	1,213.44	1,277.666	1,603.013	1,316.491

parison of the weights of all the vessels of each class will be found interesting.

			ARMORED CRUISERS.		DESTROYERS.		
	Engines, 4-cylinder triple.		Water-tube boilers.		Water-tube boilers.		
	Thornycroft. — *Missouri.*	Thornycroft. — *Ohio.*	Niclausse. — *Pennsylvania.*	B. & W. — *Maryland.*	Thornycroft. — *Chauncey.*	Thornycroft. — *Paul Jones.*	Thornycroft. — *Worden.*
	112.289	134.826	162.66	175.03	21.057	20.452	19.967
	74.527	70.982	134.19	122.7	16.655	15.547	18.953
	101.336	127.216	163.3	135.23	14.85	12.56	13.945
	34.775	33.57	47.64	50.24	5.109	4.107	5.614
	24.895	34.883	38.29	38.2	4.816	3.568	5.536
	30.151	33.872	45.56	47.55	7.876	7.761	7.658
	18.839	12.269	32.27	31.29	2.598	2.827	2.306
	24.124	21.6	28.24	28.23	2.187	1.498	2.078
	265.365	259.975	701.81	469	56.697	58.887	62.165
	92.317	101.603	96.73	171.82	13.898	17.338	12.777
	119.458	120.926	163.98	162.81	6.904	5.267	3.452
	56.56	47.642	82.39	74.22	9.125	9.493	10.665
	43	44.644	62.73	78.62	8.125	9.305	6.842
	11.321	17.899	8.26	14.82	2.434	3.272	2.647
	45.129	56.578	42.9	50.82	3.75	3.977	3.628
	46.812	51.244	71.88	63.3	6.491	8	6.627
	34.682	39.248	38	41.25	5.74	6.043	3.745
	29.855	34.013	47.6	66.41	2.046	.862	1.027
	53.126	45.302	50.21	63.29	4.185	2.157	2.934
	17.909	10.383	13.14	12.62	.348	.419	.357
	1,236.47	1,302.686	2,031.78	1,898.05	195.198	193.34	192.923
	80.021	68.068	153.28	173.26	14.522	12.395	14.55
	1,316.491	1,370.754	2,185.06	2,071.31	209.72	205.735	207.473

Table II.—ACTUAL WEIGHTS OF MAIN ENGINES ALONE, WITH CRANK SHAFTS AND BEDPLATES.

——	I. Cylinders.	II. Crank shafts.	III. Bedplates, framing and guides.	IV. Reciprocating parts.	V. Valve gear.	Total pounds.	Total tons.	Pounds per actual I.P.H.	Total machinery weight.	Size of engines.
	Pounds.	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*				*Per ct. of*	
Arkansas	37,099	4,994	19,260	7,874	8,350	77,571	34.66	45.18	13.77	17—26½—40×24
Nevada	38,111	4,955	19,115	8,354	9,548	80,083	35.75	41.23	12.93	do.
Florida	33,349	4,934	18,456	7,814	8,933	73,486	32.81	31.71	14.77	do.
Wyoming	37,447	4,929	19,344	8,591	11,299	82,110	36.66	35.31	13.82	do.
Denver	62,780	10,717	41,662	17,033	16,775	148,967	66.50	26.79	14.95	18—29—35½—35½×30
DesMoines	59,946	10,322	41,739	16,329	15,756	144,142	64.47	28.49	14.26	do.
Tacoma	68,601	10,926	44,105	17,463	16,848	157,943	70.51	30.18	15.95	do.
Cleveland	62,772	10,766	40,698	16,586	16,261	147,083	65.66	32.07	14.35	do.
Illinois	203,914	41,756	132,795	59,500	54,955	492,920	220.05	38.97	17.21	33½—51—78×48
Alabama	195,160	40,860	133,733	54,961	55,055	479,769	214.18	43.32	17.65	do.
Wisconsin	217,839	41,442	147,709	60,743	62,092	529,825	236.53	42.99	18.51	do.
Maine	241,310	45,828	178,289	76,761	55,272	597,460	266	39.27	16.59	38½—59—92×42
Missouri	251,528	57,736	188,216	77,898	55,767	631,145	281.76	40.35	21.40	34½—53—63—63×48
Ohio	302,011	54,748	224,529	75,198	78,136	734,622	327.96	46.05	23.93	35½—53—63—63×48
Fennsylvania	364,367	89,178	305,633	106,712	85,761	951,651	424.84	33.98	19.44	38½—63½—74—74×48
Maryland	392,064	76,087	253,203	112,538	85,579	919,471	410.48	33.35	19.82	do.
Chauncey	47,168	7,909	25,873	11,353	10,820	103,123	46.04	12.89*	21.96*	20½—33—38—38×22
Paul Jones	45,816	7,690	22,630	9,203	11,335	96,674	43.16	12.08*	20.98*	do.
Worden	44,727	8,129	23,885	12,575	12,401	101,717	45.4	12.71*	21.88*	23—34—37—37×22

* These figures are based on estimated I.H.P.

It will be noted by referring to the sixth column in Table II that the weights of engines for vessels of the same class agree only fairly well. In the monitors, the *Florida* is somewhat lighter than the other three. In the cruisers, the *Tacoma* is a little heavy, while in the remaining classes the weights are scattering. With the *Wisconsin* 22 tons heavier than the *Alabama*, and the *Ohio* 61 tons heavier than the *Maine*, while the *Pennsylvania* is 14 tons heavier than the *Maryland*, it would seem that a closer agreement should be reached, considering that these differences in weight are for the *main engines only.*

Taking the total weight of engines as given in column six of Table II, we have:

Class.	Heaviest.	Lightest.	Difference.	Difference in lightest.
	Pounds.	*Pounds.*	*Pounds.*	*Per cent.*
Arkansas	82,110	73,486	8,624	11
Denver	157,943	144,142	13,801	9
Illinois	529,825	479,769	50,056	10
Maine	734,622	597,460	137,162	22
Maryland	951,651	919,471	32,180	3
Destroyer	103,123	96,674	6,449	6

When it is recalled that corresponding vessels of the same class were designed for the same power, there can be no doubt but that 10 per cent. and 11 per cent. is an unnecessarily large variation in weight for similar engines.

Group V.	*Arkansas.*	*Nevada.*	*Florida.*	*Wyoming.*
	Pounds.	*Pounds.*	*Pounds.*	*Pounds*
1. Eccentrics	767.8	1,014	1,002	1,052.75
2. Eccentric straps	1,147.2	1,182	986	1,072.75
3. Eccentric rods	746	702	650	843.5
4. Links and blocks	441	472	424	589
5. H.P. piston valves	156	160	118	249
6. I.P. piston valves	324	300	236	499.5 (4)
8. L.P. valves	1,061 (P)	1,056 (S)	1,078	1,459
9. Valve stems	461	508	466	556
10. Valve stem crossheads	142	174	222	226
12. Suspension links	436.5	612	606	754
13. Reversing gear	2,667.2	3,368	3,145	3,997.25
Total	8,350	9,548	8,933	11,299
Heavier than *Arkansas*	...	1,198	583	2,949

In the *Arkansas* class the only appreciable differences are in groups I and V. In group I, the *Arkansas*, *Nevada* and *Wyoming* nearly agree, while their average is 37,552 pounds, being 4,203 pounds or nearly 2 tons heavier than in the *Florida*. As to group V, the preceding table will show where the differences lie.

Similarly in the *Illinois* class, though the *Alabama* is light in group IV the discrepancies are primarily due to group V. The weight saved on the *Alabama* in group IV is interesting as showing what saving is made by considering small items. In this case it is mainly in the connecting rods, which weigh:

Illinois, ...	26,585 pounds,
Alabama, ...	22,235 pounds,
Wisconsin, ...	27,183 pounds,

a difference of 4,350 pounds between *Illinois* and *Alabama*. This amounts to 725 pounds for each rod. The rods for all three ships are of the same length and diameter, but in the *Alabama* all unnecessary metal has been cut away and the brasses cored out and lightened. Thus almost two tons of useless metal were saved and this weight could be held in reserve to balance that amount of overweight found necessary, or desirable, elsewhere.

Turning to group V, where there is a difference of 7,140 pounds between the heaviest and lightest, the details of this group follow:

GROUP V, *ILLINOIS* CLASS.

—	*Illinois.*	*Alabama.*	*Wisconsin.*
1. Eccentrics	7,479	8,528	7,983
2. Eccentric straps	9,826	8,672	10,104
3. Eccentric rods	5,524	5,247	4,494
4. Links and blocks	2,526	2,881	3,802
5. High-pressure valves	1,199	1,132	2,174
6. Intermediate-pressure valves	2,185	2,434	3,821
8. Low-pressure valves	4,485	4,259	7,314
9. Valve stems	2,133	2,072	2,036
10. Valve crossheads	1,991	1,871	2,111
12. Suspension links	3,141	2,680	3,218
13. Reversing gear	14,467	15,279	15,033
Total	54,956	55,057	62,090

Here the difference in weight is noticeable in subjects 5, 6 and 8.

It is interesting to note that in group III the weights of *Illinois* and *Alabama* agree very closely, although the former has the Bureau type of open framing, while the latter has cast inverted Y-frame on the back.

The weights of the engines of the *Maine* class present more points of interest than any of the others. Here are three vessels with engines designed by their builders, and engines which on trial developed practically the same power, as follows:

Maine, ...	15,214 I.H.P.
Missouri, ...	15,642 I.H.P.
Ohio, ...	15,951 I.H.P.

The engines of the *Maine* are 3-cylinder triple-expansion, with inverted Y-back frames and with all valves piston valves; those of the *Missouri* are 4-cylinder triple, with the Bureau type of open framing and all valves piston, while the engines of the *Ohio* are 4-cylinder triple, with inverted Y-back frames, but with slide valves on the I.P. and L.P. cylinders.

In the two 4-cylinder engines, the *Missouri* and the *Ohio*, note how closely the weights agree, both where there is little opportunity to vary the design, as in the crank shafts, group II, and where the pistons are practically the same diameter, as in group IV. It would also seem, from a comparison of the *Maine* and *Missouri*, that for the same power the weights for reciprocating parts and valve gear are practically the same, whether the engines have three or four cylinders, while the crank shafts and framing only vary by an amount which seems to reasonably allow for the differences in length due to the four-cylinder arrangement. On the other hand, in group V, note the increase in weight due to the use of slide valves on the *Ohio*, and no doubt some of the excess in group I on this ship should be charged to them as well.

The engines of the *Pennsylvania* and *Maryland* are the largest we have to consider, and although they have the same

size cylinders and the same stroke, there is a difference in weight of about 14½ tons.

The weights of the engines, in detail, for these two ships are:

Groups.	*Pennsylvania.*	*Maryland.*	Difference.
I.—H.P. cylinder	61,897	65,505	...
I.P. cylinder	90,831	98,320	...
L.P. cylinders	193,168	207,307.8	...
Relief, drain and safety valves	2,464	3,125.6	...
Receiver pipes	13,348	12,560.8	...
Stuffing boxes	2,659	5,244	...
Total	364,367	392,063.7	27,696.7
II.—Crank shafts	89,178	76,087	13,091
III.—Bedplates	176,638	...	...
Columns, tie rods, frames, etc.	128,995	...	...
Total	305,633	253,203	52,430
IV.—H.P. pistons	3,594	3,628	...
I.P. pistons	6,788	5,534.5	...
L.P. pistons	16,728	14,498	...
Piston rods	11,180	12,638	...
Crossheads and slippers	17,292	18,723	...
Connecting rods	51,130	57,517	...
Total	106,712	112,538.5	5,826.5
V.—Valve gear	85,761	85,579	182

Here the engines of the *Pennsylvania* are about 14½ tons heavier than those of the *Maryland*, and this extra weight is largely due to the fact that, in developing the Department's design, the *Pennsylvania's* crank shafts were altered by making each of them in four lengths, and interchangeable instead of in two lengths. It is noticeable that in groups IV and V, which would naturally be unaffected by the change in crank shaft, the two engines nearly agree, while in group I the *Pennsylvania's* is much lighter. The result of this change in shaft was that each crank shaft on the *Pennsylvania* had five more flange couplings than on the *Maryland*, and to accommodate these additional couplings each shaft had to be made nearly 3 feet longer. But the change was even more far reaching, for it involved lengthening the bedplates about 4 feet 2

inches, adding two more main bearings for each shaft, besides increasing the framing. Further, the *Pennsylvania's* bedplates were 3 inches wider in the top flange and 1 inch wider in the bottom flange. An inspection of the above table shows that in this way the *Pennsylvania* not only lost the weight she had gained on the cylinders and reciprocating parts, but over 5 tons additional.

In the three destroyers given we have a fairly close agreement, at least between the *Chauncey* and *Worden*, although the *Worden* engines are somewhat different from the other two. The *Paul Jones* is light in group III, also in group IV.

GROUP III.

	Bedplates.	Columns and tie rods.	Crosshead guides.	Total.
Chauncey	13,910	8,234	3,729	25,873
Paul Jones	11,360	8,157	3,113	22,630
Worden	12,359	7,494	4,032	23,885

GROUP IV.

	H.P. piston.	I.P. piston.	L.P. piston.	Piston rods, crossheads and slippers.	Connecting rods.	Total.
Chauncey	603.5	567	1,391.5	2,961	5,925	11,448
Paul Jones	497	496	1,308	2,712	4,190	9,203
Worden	665	688	1,524	3,747	5,951	12,575

The figures in the eighth and ninth columns of Table II are not of much value for comparison, for in column eight the horsepower and in column nine the total weight varies, with no connection whatever with the weights of engines; still they show, in a general way, what may be expected from the different types.

We have now finished with the weights of the main engines proper, and have considered all of groups I, IV and V. There are still parts of groups II and III to be considered, and, deducting the subjects in these groups already included in main engines, we have remaining the following:

GROUP II.—(NOT INCLUDING CRANK SHAFTS.)

—	Line shaft.	Propeller and stern-tube shafts.	Thrust shafts.	Shaft couplings.	Coupling bolts and nuts.	Total.	Total length shafting.	Weight per running foot.
	Pounds.		*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Ft. In.*	*Pounds.*
Arkansas	11,027		2,570	646	185	14,528	135	107.6
Nevada	9,268		2,630	696	156	12,770	108–10	117.3
Florida	11,249		2,423	624	159	14,455	109–11	131.5
Wyoming	11,212		2,516	1,014	210	14,952	109–11	136
Denver	7,518	22,784	3,557	1,599	664	36,122	181–10	...
Des Moines	7,925	21,283	3,550	1,415	410	34,583	181	191
Tacoma	7,840	22,319	3,578	751	635	35,123	181– 2	194
Cleveland	7,524	21,882	3,529	721	702	34,358	181– 9	180.9
Illinois	24,350	48,637	12,378	3,332	3,931	92,628	203– 2	456
Alabama	24,500	47,570	12,762	4,048	3,033	91,913	202– 9	453
Wisconsin	23,192	49,504	12,404	3,278	3,213	91,591	202– 9	452
Maine	26,718	81,251	19,049	8,797	4,784	140,599	206– 9	680
Missouri	22,055	62,113	16,810	3,457	4,770	109,205	194– 9	561
Ohio	26,495	52,573	16,300	4,665	4,914	104,249	191	546
Pennsylvania	23,680	136,907	28,250	18,183	4,388	211,408	250– 5	844
Maryland	27,275	128,105	21,202	19,735	2,430	198,747	249–10	796
Chauncey	24,766		1,565	2,566	465	29,362	199– 1	147
Paul Jones	23,213		1,688	1,808	432	27,141	208– 6	130
Worden	27,184		1,902	3,929	1,311	34,326	...	...

In certain items in the remaining parts of group III there are great variations between the different vessels, as—

Thrustbearings:	Weight.	Difference between heaviest and lightest.
Arkansas,	6,016.4	2,735
Nevada,	3,281	
Florida,	5,370	
Wyoming,	4,911	
Illinois,	20,160	3,590
Alabama,	19,367	
Wisconsin,	22,957	
Pennsylvania,	39,473	4,160
Maryland,	35,313	
Chauncey,	4,201	1,094
Paul Jones,	3,106.5	
Worden,	3,856	

Turning to group VI, main condensers, we find that here, as elsewhere, the weights agree only fairly well. For the same power, and practically the same cooling surface, a difference of one or two tons seems too much, particularly when we find such a case as is presented by the *Maine* class, where the smallest condenser is the heaviest. The following table gives the particulars in detail:

ACTUAL SIZE AND WEIGHTS OF MAIN CONDENSERS.

	Cooling surface. One.	Inside diameter.	Length between tube sheets.	Number of tubes in one.	Weight of two.	Weight per square foot of cooling surface.	Difference in weight between heaviest and lightest.
		Ft. In.	*Ft. In.*		*Pounds.*	*Pounds.*	*Pounds.*
Arkansas	1,600	...	10- 9½	905	27,136	8.48	3,956
Nevada	1,612	...	10- 9½	913	29,898	9.27	
Florida	1,612	...	10- 9½	913	27,657	8.58	
Wyoming	1,612	...	10- 9½	913	31,095	9.64	
Denver	3,005	4- 4½	9- 2	2,004	27,556	4.59	1,155
Des Moines	2,992	4- 4½	9- 2	1,996	26,897	4.49	
Tacoma	3,005	4- 4½	9- 2	2,004	27,897	4.64	
Cleveland	3,004	4- 4½	9- 2	2,004	26,742	4.45	
Illinois	7,050	5- 8½	11- 6	3,749	53,084	3.76	2,934
Alabama	7,002	5-10½	11- 6	3,722	51,449	3.67	
Wisconsin	7,006	5-10 1/16	11- 6	3,724	54,383	3.88	
Maine	9,729	6- 3	13- 4	4,460	70,472	3.62	8,335
Missouri	9,381	6- 5	12- 3	4,680	67,539	3.6	
Ohio	9,040	6- 2	11-10	4,769	75,874	4.19	
Pennsylvania	14,329	7- 0	15- 6	5,662	102,056	3.56	4,449
Maryland	14,411	7- 5	14	6,292	106,505	3.69	
Chauncey	3,470	51 13/16	9- 3	2,292	17,643	2.54	490
Paul Jones	3,470	53¾	9- 3	2,292	17,387	2.5	
Worden	3,496	44	13	1,642	17,153	2.45	

The sixth column in this table may be found useful.

In group VII, main air and circulating pumps, we have two most important auxiliaries. There is not much opportunity for saving weight in connection with the air pumps, except

possibly in using a double acting pump, as was done on the *Missouri*.

In the case of the circulating pumps and engines, however, there are wide discrepancies. The work for these auxiliaries is almost always definitely stated in the specifications, giving the number of gallons per minute required to be pumped from the bilge, and, usually, the revolutions per minute as well. One would think that with these particulars fixed there would naturally be a very close agreement between the size of the engines and pumps, supplied by different makers, to do the same work. A glance at the following table, however, will show how the size and weight of these auxiliaries vary:

GROUP VII.—MAIN AIR AND CIRCULATING PUMPS.

	Air pumps.		Circulating pumps.		Circulating pump engines.	
	Size.	Weight of two.	Diameter of runner.	Weight of two.	Size.	Weight of two.
Arkansas	16½ × 8½	3,000	30	2,347	5 — 5 × 6	2,369
Nevada	16½ × 8½	3,089	26	2,068	5 — 5 × 6	1,576
Florida	16½ × 8½	2,989	22	1,649	5 — 5 × 5	1,649
Wyoming	16½ × 8½	3,035	27	1,611	4½ — 9 × 6	2,714
Denver	22 × 10	4,948	36	4,702	5 — 10 × 6	2,137
DesMoines	22 × 10	4,736	30	5,210	5 — 10 × 9	3,333
Tacoma	22 × 10	5,213	29	3,128	5 — 10 × 6	2,748
Cleveland	22 × 10	5,044	36	4,754	6 — 6 × 6	2,706
Illinois	2, 12—2, 25 × 18	18,824	42	7,872	11 × 8	9,038
Alabama	2, 12—2, 23 × 20	18,703	54	10,586	14 × 12	6,984
Wisconsin	2, 14—2, 28 × 18	17,728	36	4,862	8½ — 13½ × 8	6,379
Maine	2, 14—2, 28 × 18	28,121	54	10,619	7 — 14 × 12	7,705
Missouri	2, 12—2, 18 × 22	22,882	48	10,482	10 — 10 × 10	8,836
Ohio	2, 12—2, 25 × 20	19,201	38	6,339	9 — 17 × 10	10,903
Pennsylvania	2, 14—2, 35 × 18	41,661	54	12,090	11 — 19 × 12	18,525
Maryland	2, 14—2, 35 × 18	41,970	45	13,568	11 — 22 × 10	14,554
Chauncey	12 × 6	4,101	20	856	4½ × 5	856
Paul Jones	12 × 6	4,149	22	982	6 × 5	1,204
Worden	12½ × 6	4,033	20	560	3 × 4	573

Similarly the table of group VIII, propellers, will show how much difference there can be in the weights of objects supposedly very similar and which have been designed to do the same work.

GROUP VIII.

	Diameter.	Disc area.	Developed area.	Weight of two.	Pounds per square foot of developed area.	Weight for one.	
						Hub.	Three blades.
	Ft. In.						
Arkansas	9	63.62	25.6	6,720.4	131.3	970	2,190
Nevada	9-2.7	66.6	24.03	6,392	133	965	2,042
Florida	8-6	56.74	25.95	8,194	157.9	1,187	2,660
Wyoming	8	50.26	25.8	7,508	145.5	1,182	2,224
Denver	11	88.8	50	18,248	182	2,336	6,210
DesMoines	10-6	86.59	34	11,824	173	1,851	3,769
Tacoma	10	78.54	26.75	10,226	191	1,935	2,701
Cleveland	10-6	86.59	34.74	12,234	176	2,024	3,692
Illinois	16-9	220.35	84.2	52,956	314.5	8,189	16,266
Alabama	15	201.06	76	43,677	287.4	6,250	14,522
Wisconsin	15-6	188.7	68	40,921	300.8	7,702	11,024
Main	16	201.06	79	46,877	296.7	5,880	15,584
Missouri	17	216.98	73.7	54,039	366.5	8,096	17,270
Ohio	16-3	207.39	90	48,382	268.9	8,870	13,394
Pennsylvania	18	254.47	92	63,250	343.8	9,062	20,764
Maryland	17-3	233.7	83.7	63,237	377.7	8,770	20,575
Chauncey	8	48.6	24.9	4,900	98.4	...	...
Paul Jones	7-5	43.2	24	3,356	70	...	...
Worden	8	50.26	22.7	4,655	102.5	...	...

We have now finished with those parts most intimately connected with the main engines proper, and, as the groups to be considered next will be found to vary in weight even more, and for quite different reasons, it may be well here to sum up briefly our conclusions regarding the first eight groups.

Speaking generally, the first thing that strikes one is that there must be great differences in design, or in placing of machinery, to cause such differences in weight as in some cases exist. Why this is so it is difficult to see, and it can have

little bearing on the performance of the machinery when finished, for all of the vessels considered here had very close trial results in their respective classes. We are forced, then, to charge most of these differences in design either to some consideration outside of the efficiency of the machinery or to the individual ideas of those entrusted with the design of the different parts, and, from the standpoint of the machinery efficiency, neither of these considerations is worthy of weight.

The second noticeable feature is the very much closer agreement in weight between the forged and machined parts than between those that are cast. To a certain extent this must necessarily result and cannot be avoided, for castings made from the same patterns, by different foundrymen, would vary in weight due to rapping of patterns, etc. At the same time this variation would not be enough to account for the really great differences we have seen, and here as well it must be due to difference in design, such as larger and thicker ribs, heavier flanges, or a number of causes of this sort.

Proceeding with the group weights, we will take together groups X, boilers; XI, boiler fittings, and, as with water-tube boilers the subject of lagging is included in group X, for the sake of comparison we will take here that part of group XIV that includes the lagging for the Scotch boilers in the *Illinois*, *Alabama* and *Wisconsin*.

In the following table, in most cases, a comparison by subjects is of little value. It is, of course, well known that among water-tube boilers there are great differences in weight between different types. It may not always be realized, however, how great these differences are, and an inspection of the table should certainly make it clear that, among other considerations of boiler efficiency for a particular case, that of weight should not be entirely ignored. The great excess of weight of Scotch boilers over any of the types of water-tube boilers should exert very strong influence towards the use of the latter type. It should be noted that the specifications for material permit the mills a variation in the weight of boiler tubes of from 5 per cent. above to 3 per cent. below the calculated weight based

BOILERS.—GROUPS IX, X AND XIV.

Name.	Kind.	Group IX, boilers.	Group X, fittings.	Group XIV, Lagging.	Total dry pounds.	Total dry tons.	Group XVIII, water.	Total wet pounds.	Total wet tons.		Total H. S.	Total G. S.	Weight per sq. ft. of H.S wet.
Arkansas	Thornycroft.	133,609	36,528	...	170,137	75.95	18,800	188,937	84.35	4	9,368	198	20.16
Nevada	Niclausse.	124,054	96,150	...	220,204	98.31	19,760	239,964	107.13	4	8,874	220	27.04
Florida	Mosher.	79,501	32,658	...	112,159	50.07	20,880	133,039	59.39	4	9,600	240	13.85
Wyoming	B. & W.	124,313	61,513	...	185,826	82.96	27,716	213,542	95.33	4	8,800	200	21.11
Denver	B. & W.	214,495	86,180	...	300,675	134.23	46,416	347,091	154.95	6	13,200	300	26.29
Des Moines	B. & W.	215,826	85,533	...	301,459	134.54	46,398	347,757	155.25	6	13,200	300	26.34
Tacoma	B. & W.	215,494	83,982	...	299,476	133.69	46,416	345,892	154.42	6	13,200	300	26.1
Cleveland	B. & W.	215,418	81,516	...	296,934	132.56	46,398	343,332	153.27	6	13,200	300	26.01
Illinois	Scotch.	743,162	136,191	27,674	907,027	404.92	391,200	1,298,227	579.56	8	21,649	684.8	57.96
Alabama	Scotch.	752,376	135,708	38,579	926,663	413.69	313,093	1,239,756	553.46	8	21,691.5	697.5	57.33
Wisconsin	Scotch.	732,456	123,966	26,443	882,885	385.22	331,734	1,214,599	542.23	8	21,205	640	57.27
Maine	Niclausse.	1,153,074	166,796	...	1,219,870	544.58	200,920	1,420,790	634.28	24	53.345	1,353.6	26.63
Missouri	Thornycroft.	594,421	206,793	...	801,213	357.68	97,380	898,593	401.16	12	51,372	972	17.49
Ohio	Thornycroft.	582,346	227,595	...	809,941	361.58	78,624	888,565	396.68	12	53,256	924	16.68
Pennsylvania	Niclausse.	1,572,067	211,555	...	783,622	796.26	234,992	2,018,614	901.17	32	68,528	1,600	29.45
Maryland	B. & W.	1,051,900	384,874	...	1,436,684	641.38	252,000	1,688,684	753.88	16	70,944	1,600	23.8
Chauncey	Thornycroft.	127,678	31,123	...	158,801	70.89	19,166	177,967	79.45	4	17,768	315	10.16
Paul Jones	Thornycroft.	134,143	36,607	...	170,749	76.23	14,260	185,009	82.59	4	17,783	315	10.4
Worden	Thornycroft.	139,250	28,620	...	167,870	74.94	20,301	188,171	84.01	4	17,768	315	10.59

on the designed dimensions, and as a matter of fact the tendency is nearly always for such tubes to be somewhat heavier rather than lighter. Similarly in boiler and other plates there is a slight variation in weight allowed, and here also the tendency is for the plates to become heavier as the rolls wear.

In connection with the boilers it may be well to call attention to the fact that contractors frequently increase the grate and heating surface of boilers without apparently considering the question of the greater weight which such increases will invariably cause. The question in such cases should be well considered by them, for the responsibility rests entirely with themselves, and, should they deem it absolutely necessary to the fulfillment of the contract, they should take into consideration the increased weight in making their estimates.

In group XI, smoke pipes and uptakes, we have reached one of the most fruitful groups for discrepancies between the weights of sister ships.

There are often great differences between the weights of

GROUP XI.—SMOKE PIPES AND UPTAKES.

——	Uptakes.	Smoke pipes.	Total.
Arkansas	7,583	23,500	31,083
Nevada	20,726	24,800	45,526
Florida	12,534	25,551	38,085
Wyoming	5,144	20,444	25,088
Denver	19,294	40,268	59,562
DesMoines	23,152	37,064	60,216
Tacoma	21,865	38,639	60,504
Cleveland	23,486	45,614	69,100
Illinois	...	...	155,667
Alabama	...	...	135,699
Wisconsin	...	...	132,932
Maine	...	...	331,818
Missouri	...	...	267,589
Ohio	...	...	270,876
Pennsylvania	...	...	367,327
Maryland	...	...	364,699
Chauncey	...	...	15,465
Paul Jones	...	...	11,802
Worden	...	...	7,732

uptakes and also of smoke pipes, and it is usually due to the difficulty in deciding exactly where to draw the line between where uptake ends and smoke pipe begins; this, however, is a matter of detail and would in no way affect the total of the group, which is the sum of the two subjects.

Though the specifications are usually very definite on these points, shipbuilders vary greatly in their design of uptakes and smoke pipes, in respect to the size, kind, location and number of stiffeners used and even in regard to the thickness of plates.

Then, again, it is not unusual to see long and tortuous uptakes, particularly in cases where, as in the *Denver* class, they run for some distance fore-and-aft, and up through the protective deck before joining at the base of the stack. The preceding table gives the particulars of this group.

The type of boiler used will often affect the uptake weights to a considerable extent. When the gases can be taken from the adjacent sides of a pair of boilers a considerable saving in weight is made over cases in which the gases must be taken from an opening in the top, usually the full width of the boiler. An example of the lighter type is presented by the *Wyoming*. As to the smoke pipes, the following table will indicate, in a general way, how slight variations in construction affect the weight.

—	*Arkansas.*	*Nevada.*	*Florida.*	*Wyoming.*
Height	55 ft. 11½ in.	55 ft.	58 ft.	57 ft. 3 in.
Inside diameter	6 ft. 3 in.	6 ft.	6 ft. 4 in.	6 ft. 2 in.
Air space	3 in.	3½ in.	3 in.	3 in.
Inside plates	No. 10 and No. 7.	No. 10 and No. 7.	Half No. 10, half No 7.	No. 10 and No. 7.
Outside plates	No. 12; lower course, No 10.	No. 12; lower course, No. 7.	Half No. 12, half No. 10.	No. 12; lower course, No. 10.
Inner stiffeners	4—3×3 T.	4—3×5 T.	4—3×3 L.	4—3×2 L.
Outer stiffeners	4—3×3 T.	4—3×5 T.	4—3×3 L.	4—3×3 L.
Weight	23,500	24,800	25,551	20,444

Groups XII, XIII and XIV, steam and exhaust pipes, suction and discharge pipes, and lagging and clothing, often show marked differences. In these cases certain differences must exist because of different leads of piping, thickness of valves, number of valves, etc. The location of auxiliary machinery will often affect the quantity of piping, &c., to a marked de-

gree, and where possible, without sacrificing other things, such machinery should be so located as to require the least possible amount of piping.

In naval work the weight of these groups must always be very great, compared with ordinary merchant work, from the necessity of nearly always providing, in the former, cut outs and duplicate connections and attachments to permit of several ways of doing the same thing.

——	Group XII, steam and exhaust pipes.	Group XIII, suction and disch. pipes.	Group XIV, lagging.
Arkansas	28,991	30,898	5,995
Nevada	27,642	32,411	5,346
Florida	29,084	27,808	5,202
Wyoming	21,696	25,665	6,325
Denver	33,552	40,178	9,887
Des Moines	35,813	56,819	13,715
Tacoma	31,129	46,677	10,009
Cleveland	34,807	65,076	11,191
Illinois	89,769	72,987	67,132
Alabama	77,274	73,244	64,421
Wisconsin	89,556	82,740	46,641
Maine	144,438	116,807	30,682
Missouri	126,702	96,326	25,363
Ohio	105,747	100,009	40,095
Pennsylvania	184,589	141,226	18,519
Maryland	166,266	176,113	33,193
Chauncey	20,446	18,285	5,465
Paul Jones	21,247	20,882	7,335
Worden	23,824	15,382	5,810

In group XV, ladders and gratings, there is sometimes an opportunity to save weight; many ships have too much grating, and often the ladders are of a very heavy type of construction. The open-tread form of ladder, which is now permitted by most specifications, when combined with angle-iron sides, permits of a very light and strong ladder. The old cast-iron treads were heavy and cumbersome, and their use should be avoided. Some of the forms of gratings could be lightened, and there is often an opportunity to save weight in floor plates, particularly in the supports. Following are the weights of this group:

GROUP XV.—LADDERS, GRATINGS, FLOORS, ETC.

	Pounds.		Pounds.
Arkansas	17,507	*Maine*	76,939
Nevada	19,122	*Missouri*	96,415
Florida	14,333	*Ohio*	126,738
Wyoming	28,861		
Denver	32,348	*Pennsylvania*	96,117
Des Moines	39,540	*Maryland*	113,832
Tacoma	39,341		
Cleveland	41,885	*Chauncey*	8,402
		Paul Jones	8,917
Illinois	79,149	*Worden*	8,126
Alabama	79,015		
Wisconsin	81,942		

Group XVI, auxiliaries, will be given in detail. A close inspection of the table will show that the differences in weight are much greater than might be supposed. Often this is due to the fact that one builder may employ a larger auxiliary for certain work than another; there is, however, considerable difference in weight between auxiliary machinery of the same size built by different makers.

In group XVII, fittings and gear, we also find great differences in weight. The following is a partial table of this group.

GROUP XVII.—(PART) FITTINGS AND GEAR.

—	Feed and filter tanks.	Water-service pipes.	Lubricating gear.	Separators.	Feed heaters.
Nevada	4,386	1,479	1,823	2,025	2,594
Florida	4,052	1,279	1,004	2,129	3,536
Wyoming	4,929	1,337	996	1,975	3,990
Denver	12,679	2,317	1,963	1,521	6,825
Des Moines	11,630	2,111	1,415	1,058	3,420
Tacoma	12,757	3,148	2,034	2,629	3,071
Cleveland	12,449	2,598	2,381	1,252	2,508
Illinois	11,080	3,317	2,272	4,819	10,053
Alabama	9,968	3,336	3,519	7,923	11,866
Wisconsin	17,794	6,484	3,415	2,936	13,674
Maine	21,202	3,309	4,100	12,054	9,143
Missouri	19,799	5,286	3,947	6,179	11,748
Ohio	23,024	7,742	2,794	5,808	17,794
Pennsylvania	21,319	7,141	4,136	12,297	11,859
Maryland	28,484	6,151	4,007	6,537	12,609
Chauncey	1,478	1,037	741	1,279	4,163
Paul Jones	2,777	1,472	747	1,545	2,434
Worden	1,427	503	658	...	2,567

GROUP XVI.—AUXILIARIES.

——	Feed pumps.	F. & B. pumps.	W. S. pumps.	Other pumps.	Aux. condensers and pumps.	Shaft pumps.	Turning engine.	Ash hoists.	Blowers.
Arkansas	3,328	1,126	1,129	1,427	5,561	630.3	1,341	1,014	5,357
Nevada	3,186	1,510	1,091	1,412	5,590	...	1,788	1,417	4,206
Florida	3,415	914	1,022	2,429	5,566	...	1,143	979	4,866
Wyoming	4,088	1,380	1,374	1,126	4,935	745.5	2,451	1,461	3,170
Denver	9,376	5,699	2,796	2,835	9,226	373	3,472	5,632	7,411
Des Moines	8,992	4,817	2,417	2,403	9,362	542	4,391	4,604	8,064
Tacoma	9,214	5,481	2,762	2,740	8,940	729	2,838	3,134	5,120
Cleveland	9,335	5,833	2,875	2,916	9,198	724	3,495	2,418	2,557
Illinois	19,396	6,552	6,486	...	18,673	6,216	5,598	5,141	11,218
Alabama	20,260	6,761	6,628	67	14,378	3,155	8,333	4,931	10,206
Wisconsin	18,056	6,001	6,016	...	13,439	3,572	8,818	5,012	16,856
Maine	41,155	7,378	7,147	6,872	15,401	3,773	8,896	8,599	19,157
Missouri	28,856	6,334	6,379	6,018	20,196	6,785	5,441	7,266	17,590
Ohio	21,243	6,644	6,666	9,194	15,401	3,663	7,306	8,184	37,587
Pennsylvania	53,184	6,253	...	8,878	15,017	1,600	11,483	11,200	43,196
Maryland	50,175	5,859	...	8,722	12,835	1,382	7,436	8,730	36,154
Chauncey	6,036	1,991	...	...	...	...	740	1,644	4,129
Paul Jones	7,819	2,731	...	...	165.5	...	422	730	2,802
Worden	6,916	2,380	...	...	...	...	706	806	4,034

Group XVIII—Water. In this group the principal differences in weight are due to difference in size of feed tanks, different types of boilers and difference in piping.

A partial tabulation of this group is as follows:

GROUP XVIII.—(PART) WATER.

—	Water in condensers.	Water in feed and filter tanks.	Water in boilers.
Arkansas	2,200	4,160	18,800
Nevada	2,984	8,126	19,760
Florida	2,556	5,702	20,880
Wyoming	2,812	5,718	47,716
Denver	5,312	19,880	46,416
DesMoines	6,718	16,204	46,398
Tacoma	6,200	21,680	46,416
Cleveland	6,292	19,374	46,398
Illinois	14,790	20,300	391,200
Alabama	13,722	23,362	313,093
Wisconsin	16,146	17,756	331,735
Maine	18,200	...	200,920
Missouri	19,220	...	97,380
Ohio	17,536	...	78,624
Pennsylvania	24,000	59,272	234,992
Maryland	27,916	76,800	252,000
Chauncey	8,525	2,330	19,166
Paul Jones	7,082	3,532	14,260
Worden	7,546	2,320	20,301

GROUP XX.—MISCELLANEOUS MACHINERY.

This group is composed of the independent plants under the Bureau of Steam Engineering, the distilling, heating, refrigerating and air-compressor plants. Some of them agree fairly, but in others, notably the heating plant, there are great differences.

It is believed that if attention was given to the heating plants, with a view to reducing the quantity of piping used, and if the use of cast-iron radiators was avoided, coils being used, a material saving in weights would result.

In the evaporating and distilling plant effort should be made to use as few evaporators and distillers as possible, the

gain then being not only in these items but in the piping as well. On sister ships using two evaporators compared with those using three, the total capacity being the same in each case, considerable gain is apparent in favor of the smaller installation.

Group XXII is composed of the piping necessary to operate those auxiliaries not under the Bureau of Steam Engineering. Here there is bound to be more or less difference, owing to the different location of these auxiliaries on different vessels, but by care in making the pipe leads as direct as possible, and, whenever it can be done, by working from a single main, an appreciable gain can be made.

Regarding the distribution of machinery weights in different parts of the vessel, the following table, giving averages of the vessels of each class, may be found of interest:

PER CENT. OF TOTAL MACHINERY WEIGHT IN DIFFERENT COMPARTMENTS.

——	Stern.	Engine rooms.	Firerooms.	Bow.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Monitor class	5.22	37.14	53.27	4.37
Denver class	7	40.18	49.10	3 72
Illinois class	6.56	33.80	58.04	1.60
Maine class	8.34	37.27	52.86	1.53
Pennsylvania class	6.20	33.67	58.88	1.25
Destroyer class	5.77	41.08	52.76	.39

It will be noted here that the percentages in the different compartments agree very well, considering that there are always some items, such as distilling, dynamo and refrigerating plants, the location of which varies somewhat with every design. Of course, the boiler and engine-room percentages will shift somewhat between themselves, depending upon whether light or heavy boilers are used, as illustrated by the *Illinois* class with Scotch boilers.

It should be emphasized that as in naval work the machinery installation is the only element of a vessel on which a penalty is exacted for excess weight, it would seem important to give great attention to this subject of weight during the period of

design and construction. Further, from the fact that some contractors save weight in one group and some in another, it is plain that in many cases a substantial saving could be made by not losing sight of this important feature in the regular course of design of details, and that it is in the consideration of individual and small items that the best opportunity for weight saving is offered. Of course, considerations much more important than weight must be the deciding elements in design, still it should be recognized that when the machinery has been designed by the Bureau, the specifications written on that design and the contract based on the two, it is not unreasonable to infer that the machinery described is what the Department desires, so that changes involving increase in weight should be made with caution, and not unless there is some prospect of saving elsewhere an amount equal to the probable excess.

It seems to be considered, in a general way, that if machinery is heavier it is better, more durable, less likely to break down; that excess in weight means increase in strength and in endurance, and is not in itself a disadvantage, and in a sense, and up to a certain point, and considering the machinery installation alone, greater weight undoubtedly does add to the strength and to the endurance of the various parts and does make for a better installation, but it is necessary that this excess weight be placed where most needed. If a contractor wishes to add thrust collars (so increasing the length and weight of bearings), if he wishes to have two sizes of connecting rods (thus doubling the number of spares), he should consider the increased weight. If a large thrust bearing is mounted on a heavy, box-shaped sole plate, instead of on one of the usual flat type, though a great deal of weight is added, there is no gain in strength or endurance or efficiency. If cylinder barrels are carried considerably above the tops of liners, with the cylinder covers extending down inside, thus making a double wall, there is no great advantage over the lighter form, though there is greater weight. It is in such unimportant details as these that lack of consideration of weight is shown.

In the course of developing designs and of construction there are often reasons, and good reasons, why certain items should be increased in weight, but these reasons may very well have nothing whatever to do with the efficiency of the vessel or of her machinery, and a change in design is not necessarily an advantage to the Government. Could each individual case be decided on its merits there would often be no reason why the additional weight should not be employed; but each case cannot be decided on its merits, for an important point in naval work is the mutual relation of the finished weights. To illustrate: A contractor, in getting out the spring bearings for a new ship, finds that he can, with very little change, use old patterns made for a former vessel having a considerably larger shaft; this is perfectly legitimate and in merchant work would not make the least difference. Suppose, however, that, owing to the larger pattern, 250 pounds was added to the weight of each bearing and that each of the twin shafts had three bearings, this would make a total of 1,500 pounds, nearly three-fourths of a ton, in this one item alone. Such an imaginary case as this is cited merely as an example, but multiply the instances of this sort and we would have a vessel that was forced to carry around as freight, during her entire active career, an utterly useless dead weight which might very well have been utilized elsewhere to advantage. Of course, such cases as this, when taken separately in this way, seem trivial and unimportant, but it is for this very reason that they should be guarded against, as in the aggregate the weight so added amounts to a great deal.

It might be urged that in making heavy machinery a builder has paid for extra material for which he receives no compensation, and if the machinery were made heavier consistently, and always with the idea of strengthening and making more durable certain parts, there might be some foundation for this opinion. The great variation in weight, however, in similar parts, shows that it is rarely material is added because the parts are considered weak, and it is not by any means always added in desirable or necessary places. It may be that old pat-

terns can be altered and used, or that auxiliary machinery, of possibly an unusually heavy type, can be secured at particularly advantageous terms, or it may be that, due to poor design, no attention has been paid to the question of weight. In each of these cases except the last it is clear that the builder has received what might be considered an indirect compensation for the extra material adding weight to the machinery.

In all machinery it is vital to employ weight enough, but on shipboard there are too many other uses to which weight can be applied to admit of its being spread at random or unnecessarily over various parts. A comparison of the weights of similar vessels built by different contractors will often show that in parts employed under the same conditions there will be great differences in weight.

Considering such cases in which certain vessels have parts of their machinery abnormally heavy, experience has not shown that they last any better, work any better, are less free from accidents or less liable to repair than their sister ships whose weights, in these parts, more nearly agree.

The writer does not wish to be considered as saying that, as far as the *machinery installation itself* is concerned, heavy machinery does not add to the strength and endurance, for it undoubtedly does, *up to a certain point;* but a careful consideration of the question has failed to make it clear that extra weight is always so placed as to be advantageous, and, further, it must always be borne in mind that the machinery installation is only one element of the vessel, which, to be considered a success, must harmonize with the design as a whole.

TEST OF SHELL BOILERS USING THE RONEY STOKER.

By G. W. Baird, Rear Admiral (and Formerly a Chief Engineer), U. S. Navy.

The Postmaster General, to comply with the "smoke law" in the District of Columbia, determined to substitute pea coal for Cumberland coal.

The Roney Stoker had been in use in the (twelve) boilers of the Post Office Department for about ten years. It was built to burn Cumberland coal, with an inclination too steep for anthracite, but, by providing a dam at the lower edge of the grates, and making a change in the dumping plate, the manufacturers claimed the grates would be efficient with the pea coal. The manufacturers were permitted to make these changes without cost to the Government.

(1.) The object of the experiment was to determine whether or not pea coal consumed on the Roney grates in the boilers would furnish sufficient steam to do the work required.

(2.) To determine the economic efficiency of the pea coal so consumed.

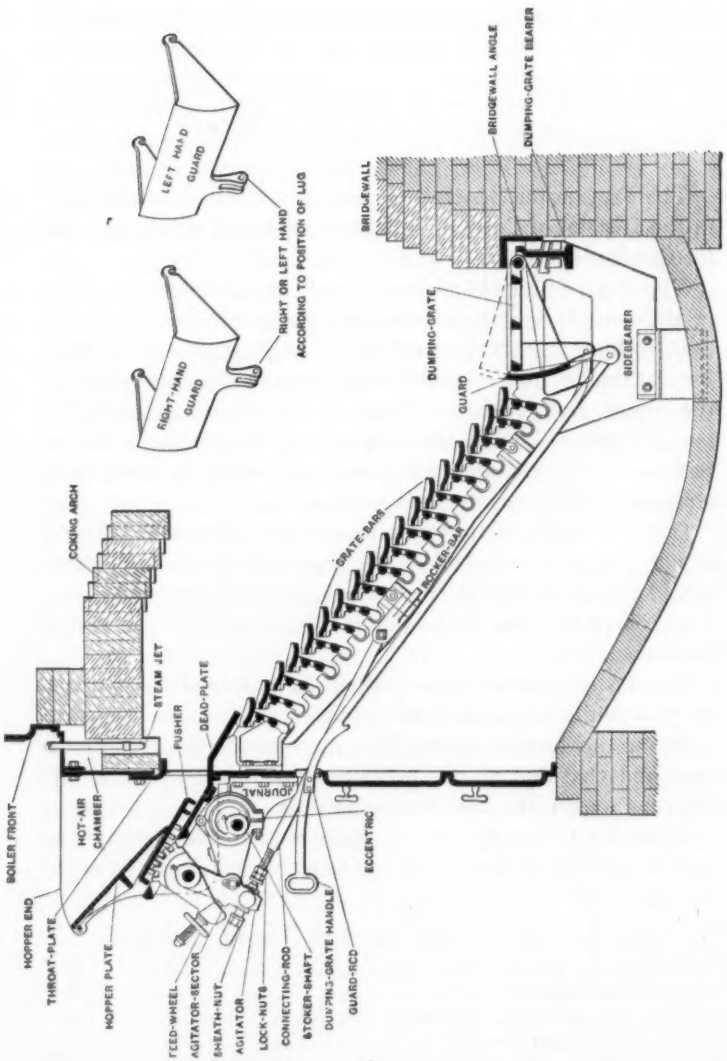
(3.) To determine what quantity of pea coal the Department would need for a year's supply.

From the records of the chief engineer of the building it appeared that during the previous year the actual consumption of Cumberland coal was 4,000 tons.

From the following tests it appears that the Cumberland coal is superior to the pea coal in economic efficiency by about 11.3 per cent.

DIMENSIONS OF THE BOILERS (SHELL BOILERS).

Diameter of the boilers, in feet	5
Length of the boilers, in feet	18
Number of furnaces (three boilers)	3
Width of each furnace, in feet	5
Projected length of the grates and dump, feet and inches	5-8
Angle of inclination of the grates, degrees	45
Total area of the grates in the three boilers, feet	85



RONEY STOKER.

External diameter of the tubes, inches	3.5
Length of tubes, feet	18
Aggregate area through tubes for draft, square feet	10.1
water heating surface, square feet	3,373.95
Ratio of heating to grate surface	39.69
grate surface to area through tubes	8.41

The feed pump is a duplex Laidlow pump, having 7.5 inches diameter of steam ends, 4.5 inches water ends and 6 inches stroke of pistons.

The feed-water heaters are of the National make, McKeesport, Pa.

The heaters discharged the feed water into an iron tank which was mounted on a platform scale, and was arranged to hold 1,300 pounds at each charge. This tank delivered into a lower tank from which the feed pump drew its supply. This lower tank was elevated sufficiently to keep a head of water on the pump, as the water was very hot.

There were two experiments, of 12 hours each; one experiment was with the pea coal; the other with Cumberland coal.

The coal and ashes were weighed on a platform scale in the fireroom.

The moisture in the steam was measured by the calorimeter which was invented by Professor C. H. Peabody, of the Massachusetts Institute of Technology, and of the form designed by Mr. G. H. Barrus. After the tests this instrument was "calibrated" for correction.

The temperature of the feed water was taken in the lower tank, close to the feed-pump suction.

The boilers delivered their steam into the main steam pipe, whence it was conveyed to the engines, which continued on their regular duty.

Each experiment was begun at noon and ended at midnight. At 2 P. M. an extra load was thrown on the engines, to ascertain if the boilers would be able to deliver sufficient steam at that load.

	Pea coal.	Cumberland coal.
Duration of experiment, hours	12	12
Total pounds of coal consumed	16,211	16,200
refuse	2,887	1,625
combustible	13,324	14,475

	Pea coal.	Cumberland coal.
Mean pounds of coal per hour	1,351	1,350
combustible per hour	1,111	1,214
coal per sq. ft. of grate per hour	15.89	17.16
combustible per sq. ft. of grate per hour	13.07	14.28
Total pounds of feed water pumped into the boilers	136,500	167,766
Mean temperature of the feed water	198.6	199.83
steam pressure in boilers, in lbs. per sq. in	102.416	102.935
lbs. of water vaporized per lb. of coal	8.42	10.35
combustible.	10.23	11.51
Mean pounds of water that would have been vaporized per pound of coal had the feed water been delivered at 212 degrees Fahrenheit	8.87	10.9
Mean pounds of water that would have been vaporized per pound of combustible had the feed water been delivered at 212 degrees Fahrenheit	10.78	12.13
Per centum of moisture in the steam	.18	1.65

The load on the dynamos (and engines) was heavier during the latter experiment, which accounts for the larger consumption of feed water.

It was not found necessary to urge the boilers at any time, though the load was above the average. The tests were made with but 25 per cent. of the boilers in use, and, though there was but little of the (exhaust) steam used in the heating system that day, it is apparent that the boiler plant is sufficient and that the pea coal will give sufficient heat for all purposes.

The purpose of mechanical stokers is not only to save labor, but, by a constant feed and by avoiding the opening of the furnace doors, the inventors and promoters often claim an absolute prevention of smoke. In this respect the Roney seems to be quite as good as any. None of them, however, have diminished the production of smoke sufficiently to satisfy the laws in the District of Columbia; the result is that the Executive Departments are preparing to burn anthracite. The cost of anthracite, compared with that of semi-bituminous coal, is a serious consideration. The pea coal and the buckwheat coal, however, are much cheaper than the furnace size of anthracite, but are less efficient pound for pound. But have the advantage over all bituminous coals of being practically smokeless.

CONTRACT TRIALS OF THE U. S. GUNBOAT *PADUCAH.*

The *Paducah* is one of the two gunboats provided for in the Act of Congress approved July 1, 1902.

The contract for this vessel was awarded to the Gas Engine and Power Company and Charles L. Seabury & Company, Consolidated, of Morris Heights, New York, on July 6, 1903, the price being $355,000, and time for delivery twenty months. This time was later extended to twenty-four months.

As a detailed description of the *Paducah* and her sister ship, the *Dubuque* appeared in the last issue of the JOURNAL (pp. 589 to 629), the principal dimensions only will be given.

The *Paducah* is a sheathed gunboat of 1,080 tons displacement, with two masts, two funnels, flush main and gun decks and an interrupted berth deck. The principal dimensions and characteristics of the hull are as follows:

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	174
on L.W.L., feet	174
over all, feet and inches	200- 9
Beam, extreme, feet	35
at L.W.L., feet	35
Ratio of length to beam	4.971 to 1
Depth, molded, main deck, feet and inches	22- 9⅛
Draught, normal, mean, feet and inches	12- 3
mean on trial, feet and inches	12- 3
Displacement, normal draught, tons	1,085.3
trial, tons	1,082.5
per inch at L.W.L., normal draught, tons	10.51
Area midship section to L.W.L., normal draught, square feet	348
L.W.L. plane (draught 12 feet 3 inches), square feet	4,470.7
Wetted surface, square feet	6,900
Center of gravity of L.W.L. plane abaft M.S., inches	3
buoyancy above bottom keel, feet and inches	7- 3
forward of M.S., feet and inches	2-11½
Transverse metacenter above center of buoyancy, feet and inches	9- ⅞
Longitudinal metacenter above center of buoyancy, feet	194
Coefficient of fineness, block	.509

Coefficient of fineness of M.S.	.812
L.W.L.	.732
Cylindrical coefficients, longitudinal	.627
vertical	.904
Number of frames	116
Number of compartments	51

ARMAMENT.

Six rapid-fire 4-inch guns; four rapid-fire 6-pounders; two rapid-fire 1-pounders; two Colt automatic.

The ship is propelled by twin outward-turning screw propellers driven by a pair of three-cylinder triple-expansion engines, with cylinder diameters of 9 inches, 15¼ inches and 25½ inches in diameter and a common stroke of 21 inches.

Steam is generated by two Babcock & Wilcox marine boilers under a gauge pressure of 265 pounds per square inch, intended to be reduced to 250 pounds (maximum) at the engine.

The boilers have a total of 4,159 square feet of heating surface and 99.78 square feet of grate surface.

The vessel has neither protective deck nor armor of any kind.

The closed-fireroom system of forced draft is used; the air pressure being supplied by two 40-inch fans, each driven by a 3×3-inch twin-cylinder simple engine.

The propellers are of the three-bladed, true-screw, solid-wheel type, having a mean pitch of 6 feet 10 inches; the blades are bent back 8 inches at the tips.

STANDARDIZATION TRIALS.

Runs for the purpose of standardizing the screws were made over the measured-mile course in Narragansett Bay on August 15, 1905; these were commenced at 8·00 A. M. and finished at 10·40 A. M. The weather was thick, rainy and squally with a smooth sea.

The draught taken before and after the runs was as follows:

	Before.	After.
Forward, feet and inches	12- 6	12- 6
Aft, feet and inches	12- 2	12- 1½
Mean, feet and inches	12- 4	12- 3¾
Corresponding displacement, tons	1,093	1,091

The data for these tables will be found on tables I and II.

Table 1.—U. S. S. *PADUCAH.*

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., August 15, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines. To make 1 knot. Starboard.	Revolutions of main engines. To make 1 knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	Starboard engine. I.P. cylinder.	Starboard engine. L.P. cylinder.	Starboard engine. Total.	Port engine. H.P. cylinder.	Port engine. I.P. cylinder.	Port engine. L.P. cylinder.	Port engine. Total.	Total both main engines.	Auxiliaries in operation (estimated)	Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1–N	7–23.8	874.15	950.45	118.18	128.50	123.34	8.111	19.31	26.49	19.13	64.93	23.98	32.24	35.36	91.58	156 51	25 15	181.66
2–S.	6–39 5	1,017.95	1,017 6	152.88	152.84	152.86	9.011	47.22	55.79	38.77	141.78	28.97	43.33	53.19	125.49	267.27	26.18	293.45
3–N.	6–00 9	982.8	986 75	163.39	164 05	163.72	9.975	63.65	71.2	55.1	189.85	54.05	63.79	77.45	195.29	385 14	25.62	410.76
4–S.	6–51	973.7	979.65	166.44	167.46	166.95	10.256	68	76.51	58.37	202 88	57.47	64.46	65 96	187 89	390.77	27.24	418.01
5–N.	5–58.2	103.8	1,039 6	173.87	174.14	174	10 05	72 74	81.6	60.98	215.32	77 97	99.07	85 97	263.01	478.33	30	508 33
6–S.	5–01	1,002	1,004 6	199.73	200 25	199 99	11.96	118.80	133 9	112.62	365.32	114.53	137.32	126.95	378.8	744.12	31.45	775.57
7–N.	5–12.8	1,099.95	1,100.85	210 99	211.15	211.07	11.509	128.94	158.63	147.99	435.56	131.81	166.24	157.22	455.27	890.83	30.39	921.22
8–S.	4–24.7	1,002.7	1,001.5	227 28	227 02	227.15	13.6	166.02	209.54	206.94	582.5	160.27	207.65	218.04	585 96	1,168.46	34.07	1,202.53
9–N.	4–52.7	1,172.8	1,173 8	240.41	240.62	240.51	12.229	194.47	242.91	235 82	673 2	170.65	246.57	276.53	693.75	1,366.95	37.31	1,404.26

Table 1-A.—STANDARDIZATION TRIALS U. S. S. *PADUCAH.*—HORSEPOWER OF AUXILIARIES.—AUGUST 15, 1905.

No of run.	Circulating pump.	Main feed pump.	Auxiliary feed pump.	Bilge pump.	Flushing pump.	Two blowers.	Dynamos.	Steering engine and hotwell pumps.	Aggregates of power.
1	S. 5.2, P. 4.1	3.6	...	1	0.5	.55	12.15	1	29.1
2	S. 6.3, P. 3.9	3.5	...	1	0.5	.56	12.32	1	28.08
3	S. 4.7, P. 2.2	5.1	...	1	0.5	.7	12.15	1	26.35
4	S. 4.7, P. 2 9	5.8	...	1	0.5	.62	13.02	1	28 54
5	S. 5.9, P. 2.8	6.2	...	1	0.5	2.1	12.84	1	31.34
6	S. 6 5, P. 2.7	7.75	...	1	0.5	1.3	12.91	1	32.66
7	S. 4.7, P. 2.3	8	...	1	0.5	2	12.67	1	31.17
8	S. 4.7, P. 3.5	8.25	...	1	0.5	4.9	12.84	1	35.69
9	S. 4.7, P. 3.8	10.6	...	1	0.5	5.1	12.84	1	39.14
	S. 5.3, P. 3.1	...	...	...	...	...	...	...	...
Averages.....	4.2	6.53	...	1	0.5	2.047	12.637	1	...

Table No. 2.—U. S. S. *PADUCAH.*

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., August 15, 1905.

Number of run.	Time at beginning of run A. M. (approximate).	Which engine.	Steam pressures.					Mean effective pressures.				Vacuum, inches.	Cut-off in cylinders from beginning, H. P., I. P., L. P.	Temperatures.							Revolutions or double strokes.								Air pressure, in inches of water.	Pyrometer, base of after stack.
			In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In first receiver, absolute.	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	L.P. cylinder.	Referred to low pressure.			Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Circulating pump.	Main-feed pumps.	Dynamo engine.	Hotwell pump.	Sanitary pump.	Bilge pump.	Auxiliary condenser pump.	Fireroom blowers.*		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	8–05	St'b'd	182	158	63	23	7.5	25	11.7	3	10.18	24.8																		
		Port	185	150	75	25	9	28.55	13.1	5.1	13.2	25		65	90	100	67	76	94	162	415	21	404	48	20	52	...	150	0	300
2	8–22	St'b'd	222	150	102.5	25	10.4	47.25	19.05	4.7	17.19	25.5					66	80	97	162	380			42				153		
		Port	225	215	107	35	12.5	29	14.8	6.45	15.28	25.2		65	87	100	66	80	97	162	438	20	404	40	24	52	...	151	0	340
3	8–44	St'b'd	222	213	127	40	12.3	59.5	22 75	6.25	21.54	25.5					66	84	104		376			44				155		
		Port	228	212	125	40	14	50.4	20.3	8.75	22.06	25.5		68	90	105	66	91	91	162	400	30	408	28	11	52	...	210	0	375
4	9–05	St'b'd	260	248	138	43	13.2	62.5	24	6.5	22.59	25.3					68	85	98		308			20				215		
		Port	265	236	120	40	12.5	52.5	25.3	7.3	22.64	25.8		62	90	105	68	92	90	176	400	34	408	48	12	60	...	200	0	400
5	9–22	St'b'd	271	250	138	44	13.5	64	24 5	6.5	22.95	25.3					66	83	116		332			19				210		
		Port	275	260	165	55	15	68.5	29.7	9 15	27.99	25.5		64	90	109	66	91	90	162	430	36	408	32	13	60	...	450	0	495
6	9–42	St'b'd	271	245	197	64	19.5	91	35	10.45	33.93	25.5					68	91	95		328			15				460		
		Port	274	242	189	68	19	87.5	35.8	11.75	35.06	...	Full.	63	90	109	66	100	92	164	441	39	409	20	13	60	...	350	0	410
7	10–03	St'b'd	260	248	207	73	22.2	93.5	39.25	13	38 26	25.4					67	96	110		326			13				310		
		Port	265	238	190	70	21	95.5	41.1	13.8	39.96	25.5		62	91	111	66	107	96	145	400	40	408	16	12	60	...	450	.15	460
8	10–24	St'b'd	275	255	247	93	29	111 75	48.13	16.88	47.51	25.5					67	105	118		318			18				420		
		Port	277	255	227	95	29	108	47.75	17.8	47.83	25.2		61	92	111	66	104	102	148	400	41	408	16	12	64	...	550	.15	510
9	10–43	St'b'd	281	258	264	106	34	123.75	52.75	18.18	51.9	24.7					66	114	128		360			13				550		
		Port	288	260	257	105	34	108.5	53.5	21.3	53.43	24.6		61	93	107	66	113	111	150	400	58	408	36	12	60	...	970	.75	490
Averages		St'b'd	248	...	...	...	...	...	...	...	...	...		...	...	...	...	...	...	...	372			30				970		
		Port	254	...	...	...	...	...	...	...	...	...		...	...	106	..	...	...	...	...	...	...	...	...	...	...	387	...	...

*On all runs forced-draft blowers ran very irregularly.

THE FOUR-HOURS FULL-SPEED TRIAL.

The contract full-speed trial was commenced at sea, at 12·30 P. M. on August 16, 1905, and finished at 4·30 P. M. The weather was squally and rainy, with a strong breeze from the northwest and a moderate sea.

The working of the machinery, both main and auxiliary, during the trial, was very satisfactory. No water was used on any bearings and all ran cool.

The boilers steamed freely and furnished easily the amount of steam required to exceed the contract speed of 12 knots per hour by nearly one knot.

The draft before and after the trial was as follows:

	Before.	After.
Forward, feet and inches	12– 5¼	12– 5
Aft, feet and inches	12– 1	12– ¼
Mean, feet and inches	12– 3⅛	12– 2⅝
Displacement, tons	1,087	1,078

A summary of the engine-room data taken during the trial follows:

PERFORMANCE ON PRELIMINARY TRIAL, AUGUST 16, 1905.

Trial began	12·30 P. M., August 16, 1905.
Duration of trial, hours	4
Steam pressure at boilers (per gauge), pounds	242.8
H.P. steam chest, S. engine (per gauge), pounds	223.6
P. engine (per gauge), pounds	218.2
1st receiver, S. engine (absolute), pounds	94.1
P. engine (absolute), pounds	103.6
2d receiver, S. engine (absolute), pounds	30.1
P. engine (absolute), pounds	32.9
Vacuum, inches of mercury S. engine	25.66
P. engine	25

MEAN EFFECTIVE PRESSURES IN CYLINDERS IN POUNDS PER SQUARE INCH.

Main engines:	
Starboard	H.P., 112.34; I.P., 49.41; L.P., 17.43
Port	H.P., 92.08; I.P., 51.06; L.P., 20.49
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard	48.58
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	49.77

REVOLUTIONS PER MINUTE.

Main engines	starboard 228.58; port 230.09
Pumps, air	starboard, attached; port, attached.
Pumps, circulating	starboard 393.1; port 396.4
feed, double strokes per minute	56.1
auxiliary, double strokes per minute, water service	14.2
bilge	51.6
Hotwell	starboard, 29.8; port 27.1
Blower engines	966
Speed of ship, in knots per hour	12.823
Slip of propeller, in per cent. of its own speed, based on mean pitch,	starboard, 16.76; port, 17.35
Air pressure in firerooms, in inches of water	.37
Temperature outside air, degrees Fahrenheit	59
fireroom, working level, degrees Fahrenheit	100

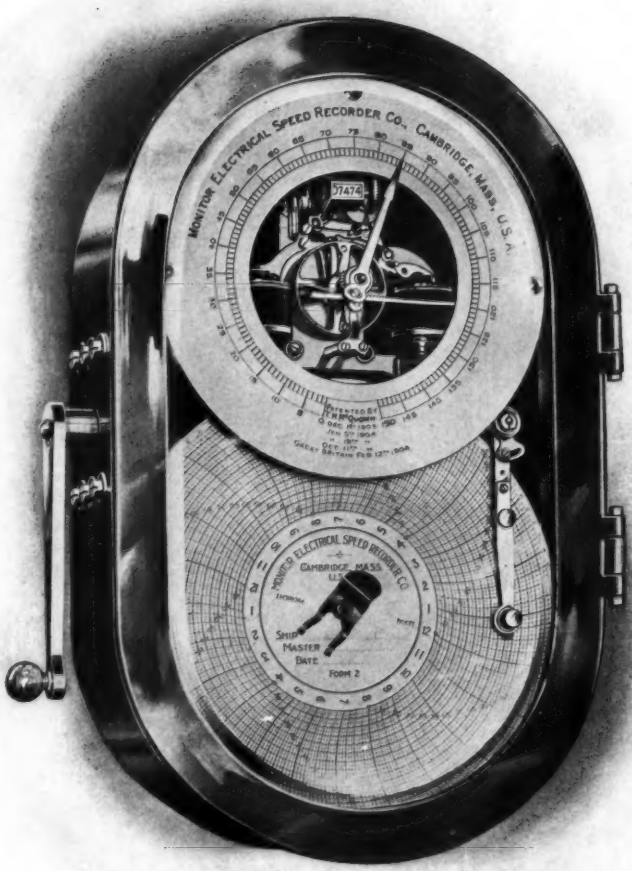
INDICATED HORSEPOWER.

Main engines:	
Starboard	H.P., 167.85; I.P., 216.34; L.P., 214.96; total, 599.15
Port	H.P., 138.49; I.P., 225.04; L.P., 254.37; total, 617.90
Air pump	starboard engine, attached; port engine, attached.
Circulating pump	starboard engine 4.55; port engine, 4.5
Feed pumps, main, one only installed	19.6
Bilge pump	1
Water-service pumps on sanitary service	.5
Ventilating engines	for ship, electric; for engine rooms, electric.
Blower engines for forced draft in firerooms (2), total	5.5
Dynamo engines	13.
Collective of all main engines	1,217.05
main engines, air, circulating hotwell and feed pumps	1,246.7
and auxiliary engines in operation during trial	1,267.7
all machinery during trial, per square foot of G.S.	12.705
H.S.	.3048
Main engines, air, circulating, hotwell and feed pumps, per square foot of G.S.	12.494
Main engines, air, circulating, hotwell and feed pumps, per square foot of H.S.	.2997

COAL.

Kind and quality	Pocahontas hand-picked, excellent.
Pounds per hour	2,698
Pounds per hour, per I.H.P., collective, all machinery in operation	2.129
Pounds per hour, I.HP., collective, main engines, air, circulating, hotwell and feed pumps	2.164
Pounds per hour, per square foot of G.S.	27.07
H.S.	.648
Cooling surface, square feet per I.H.P.	1.071
Heating surface, square feet per I.H.P.	3.28

XU



MONITOR ELECTRICAL SPEED RECORDER.

NEW ELECTRICAL SPEED RECORDER.

By Lieutenant H. C. Dinger, U. S. N., Member.*

The need of an instrument that will indicate accurately at a glance the rate of speed for an engineer located at some distance has long been felt, and numerous attempts at securing an instrument to perform this service have been made.

The Monitor Speed Recorder, a description of which follows, is designed to supply these wants in a simple, practical and feasible manner.

An instrument of this character is of special value to the officer handling a man-of-war in fleet maneuvers. In order to control the speed and movement of vessels engaged in evolutions it is most important to know accurately, at any time, just what speed the engines are making and have just been making, and also to know just when a certain ordered speed is being made.

Besides the speed of the engines, it is important to know the speed the vessel is making at any time without going to the trouble and delay of reading the log between two time intervals. The possession of an instrument that will indicate at any moment the speed of the vessel will be of great help in aiding the gunnery officer to know what orders of lateral deflection of gun's sights to allow when in action or at target practice.

The following is a specification in detail of the working and adjustment of the Monitor Electrical Speed Recorder:

On Figure 1 are shown the principal parts of the Monitor Electrical Speed Recorder.

This instrument is used to show and record the speed of various objects, such as ships by their logs, marine engines, stationary engines, railroad trains, and the velocity of the wind.

* From notes furnished by Monitor Electrical Speed Recorder Company.

For brevity, we will call the object of which the speed is to be shown the "engine."

The principle of this indicator is this: On the "engine" is an electrical make-and-break that operates a magnet C in the indicator. The frequency of the beats of the armature C1 is governed by the speed of the engine.

The armature C1, carried on a lever C3, by means of a pawl C8, drives a ratchet F3. It is obvious that the rate of speed at which the ratchet F3 is advanced varies with the speed of the "engine."

The "engine" make-and-break works continuously while the engine runs, but the advancing of ratchet F3 is interrupted every thirty seconds by the releasing mechanism which throws the pawl C8 and D7 up off the ratchet F3, which immediately is returned to its starting place by a spring in its hub.

The release is made so that pointer F is left pointing at the speed shown for the last thirty seconds. The release being completed, the advancing of ratchet F3 begins and continues for the next thirty seconds, when another release takes place, and so on.

The hand points at the speed shown for the last full thirty seconds, unless by the speed being increased it is being advanced to a still higher indication.

For ships' logs a one-minute period is used instead of the thirty-seconds period for other purposes.

INSTALLING INDICATORS.

Printed instructions and cuts for installing indicators for various purposes are sent with the indicator, but it should be noted here that it is very important that sufficient electric current be used to be sure of there being no lack of power to drive the magnets. If dry cells, wet cells or storage batteries are used, ascertain the number that will do the work, then double that number in series; this allows for the decrease of strength of the batteries, and then by the diagrams of installation will be seen how the lifetime may be greatly prolonged by wiring the batteries in multiple series. If the current is

Fig. 1.—MONITOR ELECTRICAL SPEED RECORDER.

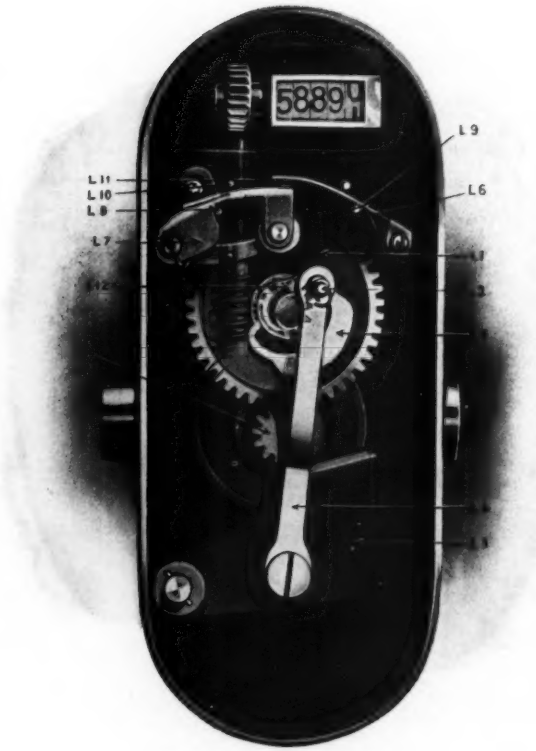


Fig. 2.—Taffrail Log.

taken from a dynamo, from 8 to 12 volts for one indicator, and not more than 500 feet of wire No. 18 B. & S. gauge. If a greater length of wire is used a higher voltage may be necessary. For two indicators to one "engine" a current of from 16 to 30 volts is advisable, according to the length of the wire.

The current should be at least 50 per cent. stronger than the amount that will just drive the magnet firmly.

Purpose of Each Part.—The work of each part is as follows: Magnet C, actuated by the "engine," operates lever C3, and by means of a pawl C8 advances ratchet F3 according to the speed of the engine.

Pawl C8 has a 1-shaped projection C9, by which it is lifted from the ratchet F3 when D7 is lifted. This projection C9 forms a second face, which engages a tooth of the ratchet ahead of the one engaged by C8; this gives double wearing surface, for F3 is driven by two teeth at a time.

At the top of lever C3 is a weight C4 which retards the quickness of the stroke given to the ratchet F3, and also reduces the noise made. Running directly back from C4 is a link C5 which prevents any severe concussion (such as from gun fire on ship board) from causing weight C4 to throw pawls C8 or K1 off their respective ratchets.

The Releasing Mechanism.—The clock electrical make-and-break, which we call the clock switch, consisting of parts A to A9, operates the magnet D, which causes the release every thirty seconds.

On the seconds arbor of the clock is a two-pointed cam A which revolves with the seconds arbor. The clocks are designed so that the seconds arbor revolves 59 times in an hour, that is, each revolution of the arbor is one minute and $\frac{1}{59}$ part of a minute. The one minute is the time used in advancing ratchet F3 (that is two periods of 30 seconds each) and the $\frac{1}{59}$ part is the time taken to make the release.

Resting on cam A are two levers A1 and A2, so adjusted that A1 falls two ticks of the clock before A2. When A1 falls it closes an electric circuit at A7 which operates magnet D, which causes the release (that is, the lifting of the pawls

from the rachet). When A2 falls this contact is broken and that release completed.

Armature D1 is given its upward movement by a flat spring D2; this upward movement is stopped by a pin D4 projecting back under the main frame.

When the contact is closed at A7, magnet D draws down armature D1, which is carried on lever D3; this draws down post D9, which is carried on the outer end of lever D3.

This post engages the tails or projections of pawls D5 and D7.

Pointer F and ratchets F1, F2 and F3 are all carried on one center post.

Ratchet F1 is mounted solidly on a sleeve with pointer F and pinion H, so that the three revolve together.

Ratchet F2 is a kind of an idler between the driving ratchet F3 and the pointer ratchet F1.

Ratchets F2 and F3 are turned backward by springs in their hubs, and ratchet F1 is turned backward by a spring in the hub H10 of the recorder arm H3 by means of a rack H2 and pinion H; so the ratchets F1, F2 and F3 will all turn backward when the pawls permit.

When the pawls are in their normal position post D9 holds pawl D6 away from ratchet F2, and pawl D5 is holding ratchet F1 and pointer F at the last indication, and pawl C8 is advancing ratchet F3.

Ratchets F1, F2 and F3 each have engaging pins. Ratchet F3 has a pin that projects through on each side, the inner end engaging screw F4 when F3 is turned back to its starting place.

Ratchet F2 has a pin that projects on each side, the inner end engaging the outer end of the pin in F3 and the outer end engaging the pin in F1. The pin in F1 projects inwardly only.

By this arrangement of the pins F1 is carried ahead of F2, and F2 ahead of driver F3, and by screw F4 they are all prevented from being turned backward beyond their starting place, which is against screw F4.

The means by which pointer F is left, indicating the speed shown for the last full thirty seconds, while the driving ratchet returns to the starting place, from which it advances for the next thirty seconds, are as follows: When the clock switch closes the contact at A7 magnet D draws down post D9; as D9 starts down, pawl D6 is allowed to engage ratchet F2; this is done in the first half of the downward movement of D9; then in the last half pawls D5 and D7 are thrown up off their ratchets by post D9 engaging their tails, and as the lifting of D7 also lifts C8, ratchet F3 is left free and is at once returned to its starting place against screw F4; also ratchet F1 is held at the new indication by its pin engaging that of F2, which is being held securely by pawl D6.

When the contact at A7 is broken and D9 lifts, it first permits pawls C8, D7 and D5 to engage their ratchets; then in the last half of the upward movement it lifts D6 from F2. F2 at once turns backwards until its pin engages that of F3, then it is carried along ahead of F3 until another release commences.

In the cut the post D9 is shown in the midway position, going either up or down. In this position all the pawls—C8, D5, D6 and D7—are in engagement with the ratchets.

As pawl D5 engages F1 before D6 has released F2 it is obvious that pointer F will be left pointing to the new indication.

The zero switch is a device for cutting the clock circuit when the pointer drops to zero, and so automatically saves the battery. The zero switch is used only on railroads and stationary engines that are stopped frequently; for ships' logs, marine engines and anemometers a hand-operated cut-out beside the indicator is preferred.

On the ratchet F1 is a pin B. When the pointer drops to zero, B engages lever B1, and by a slight movement breaks a contact at B3; this contact is in the circuit from the clock switch to the magnet D.

B4 is an insulated lug at which the heavy wire and light spring wire are connected.

The cyclometer or revolution counter consists of parts K to K7. On the upper end of the lever C3 is a second pawl K1 which drives a ratchet K. A pawl K2 carried on an adjustable lever acts as a check pawl. Ratchet K is driven forward continuously while the engine runs; it is not interrupted by the "release." To guard against K being stopped in case the clock should stop and so fail to make the release (in which case the pin F3 would make almost a full turn and lodge against the upper side of screw F4 and stop ratchet F3), at F5 two teeth are cut from F3 at a point that will come to pawl C8 just before the pin of F3 lodges against screw F4, so the pawl C8 works freely on this space and keeps the counter K7 running accurately regardless of the rest of the mechanism; this is very important in case of a ship's log when the counter shows the distance traveled.

Ratchet K drives counter K7 by means of a pair of bevel gears K4 and K5. The proportion of these gears varies according to the purpose for which the indicator is intended.

Used as a ship's log, this counter shows nautical miles to one-tenth of a mile, the right-hand figures showing tenths; also on railroad trains and anemometers it shows miles to one-tenth of a mile; but with a marine or stationary engine the total number advanced must be multiplied by ten to get the revolutions made. Applied to side-wheel steamers that never exceed 37 turns a minute, the hand and dial show the turns of the moment to one-fourth of a turn, and the counter shows the revolutions made without multiplying.

The Recorder.—This consists of the record carrier H11 revolved by the clock, and a clip to clamp the records to H11. A table H8 under the pen H7 which is carried by a spring arm H6; H6 is carried rigidly on lever H4 which works rigidly but is adjustable to lever H3.

At the upper end of H3 is pivoted a rack H2 actuated by pinion H. Rack H2 is held to its place with pinion H by a grooved roller carried on an arm H1.

A spring in the hub H10 turns H3 in the direction of the arrow.

As pointer F advances, pinion H and rack H2 swing lever H3 to the left, and so swing pin H7 to the right on a line of the record corresponding to the figure on the dial indicated by the pointer, and as the clock revolves, the record mark made by the pen shows the rate of speed at any given time.

ADJUSTING THE PARTS.

The Clock Switch.—If a thirty-seconds period is used, cam A will have two points as shown, and, as it is necessary that each period be just thirty seconds, care must be taken not to bend the point of the seconds arbor, for if the two points of A are not true, and one period should be made longer than the next, the result would be that every other indication would be too high and the alternating ones too low.

If a two-point cam is used, lever A2 should fall just two movements of the clock balance (one each way) after arm A1. To make this adjustment accurately, just before A1 drops put your finger lightly on the clock balance, get the right movement so that as you rock it back and forth the escapement will advance with each movement; when A1 drops count the movements until A2 drops; it should be just one movement each way. With a one-point cam A and the one-minute period there should be just two movements each way from the falling of A1 to that of A2.

The adjustment of the lever A3 is mostly perpendicular, and adjusts the distance A1 drops, which should be about half way from the top to the bottom point of A as shown in the cut. If A1 should drop to the bottom of A a good contact at A7 would not be obtained.

The adjustment of the lever A4 is mostly horizontal, and adjusts the length of time between the falling of A1 and that of A2. A8 and A9 are springs that cause the arms to drop, and also serve as conductors of the current of the clock circuit, and so must be insulated from their pivots, and must not touch any part of the clock framework.

Levers A1 and A2 are held to their pivots by split collars.

These must not be set too closely to the sleeves, for A1 and A2 must be perfectly free and drop quickly.

A5 and A6 are lugs at which springs A8 and A9 are connected to the larger conducting wires.

The contacts at A7 are two pieces of iridio platinum, one of which is cut V-shape. These should come together centrally each way.

Adjusting the Zero Switch.—The zero switch has iridio platinum contacts at B3, the same at the clock switch at A7.

Lever B2 should be set so that when F drops to zero the contacts at B3 will separate about $\frac{1}{32}$ of an inch. The spring on B4 should be just strong enough so that F1 will overcome it when released, say at 2 or 3 on the dial.

Adjusting the Driving Mechanism.—The movement of lever C3 is adjusted by the rubber-tipped bumpers C6 and C7.

First set C6 so that pawl C8 will drop over one tooth and go about one-third way up the next; then set up the check nut *tight*. Then, as the spring C2 advances lever C3 and ratchet F3, bumper C7 should be adjusted so that pawl D7 will fall over one tooth and go about one-third way up the next; then set up this check nut *tight*. Armature C1 should not strike magnet C by a space twice the thickness of spring C2.

Link C5 must work freely, but not permit much side movement of lever C3.

Adjusting the Releasing Mechanism.—Pawl D6 should be formed so that post D9 will hold it about one-sixteenth of an inch from ratchet F2, also so that in the descending it will strike the middle of a tooth as shown in the cut, and not on or just behind the point.

When pawl D5 descends, it, too, should strike in the middle of a tooth, not on or near the point. This on a steady rate of speed permits pointer F to drop back one-half a point as D9 comes down and one-half a point as it goes up at the end of the release. This backward drop of one point is compensated for at the start, where it will be seen by the section of dial shown that there are only four spaces from 0 to 5, but five

short spaces between each of the other long graduations, so the hand really starts at 1 instead of 0.

The adjustment of the pointer is made by screw F6.

Adjusting Ratchets F1, F2 and F3.—When F3 and F2 are put to place, see that screw F4 is removed and the hub springs set properly to place; their inner ends should hold securely in the keyway of the center post and the outer ends hooked through the slots in their hubs. See that the springs are put in so as to turn the ratchets backward, then turn the ratchets forward until the pins have passed the hole for F4 twice, then put screw F4 firmly to place; let F3 and F2 come back against F4, then put on ratchet F1 with its pin just ahead of that of F2. Put on nut F7, letting the ratchets play just enough to work freely.

The spring used in F3 is similar to a watch main spring 7 wide, 12 strong.

The spring in F2 is much lighter, being .025 ($\frac{1}{40}$) of an inch wide and .005 of an inch thick.

Adjusting the Counter.—The adjustment of the pawl K1 is made with that of C8. Check pawl K2 should be set so that K1 and K2 have equal clearance over their teeth.

Post K3 is to make it impossible for the ratchet K to bounce ahead an extra tooth; to do this K3 must be set so that K1 will lack about one-hundredth of an inch of striking K3. After setting K3 with lever C3 in its forward position as shown, try advancing ratchet K with a gentle pressure direct on K. If K3 permits K1 to raise more than half a tooth then K3 should be set still closer to K1.

By a clamp K6 the counter K7 can be adjusted towards a gear K4 so the gears will engage but will not bind in the least. Never put any oil into the counter.

Adjusting the Recorder.—The grooved roller on H1 should keep rack H2 in easy but sure engagement with pinion H. See that it does not bind at any point of the movement of pointer F.

The spring in the hub H10 is similar to a watch main spring 7 wide, 7 strong. To give this spring its proper tension,

see that the nut H10 is up tight, and set screw H9 slack, then give nut and post H10 two full turns in the direction of the arrow and hold them there by setting screw H9 up *tight.*

To adjust the pin H7 set down screw H12 until the point of the pen just touches the table H8 all along its course; H8 should be perfectly level; then put a record dial on plate H11 and have the pointer F at some advanced position, say at 30; then by screw at H5 set the pin so it will be on the corresponding line on the record. When the pointer F is at zero rack H2 should not project more than one tooth to the left of H.

If the pen marks too heavily, and it is not due to too much pressure from screw H12, take off the pen cap and insert the end of a small wire or toothpick and gently press the little brass marker out of the pen point a little farther.

Wind the clock once a week; wind both springs. Change the record dials every twenty-four hours. To take up the back lash of the gears the record should be set to the right time by a slight backward movement after the clamp is on. The movement is upward on the pen side.

The pen should be filled once in ten days or two weeks. Do not detach the pen, but simply spring it back an inch or so; remove the cap, fill the pen, using a small dropper, and replace the cap. Use only the ink supplied with this style of pen.

Figure 2 shows the regular taffrail log, make-and-break.

L is the driving gear and L1 the driven. The proportion between these gears varies according to the highest speed at which the engine is to be run, also according to whether the speed is to be indicated once a minute or every thirty seconds. In the cut, L and L1 are equal, and would be used on an engine not exceeding 150 revolutions per minute, the indication being shown once a minute.

For an engine under 150 revolutions per minute, showing the speed every thirty seconds, L would be twice the size of L1.

For an engine under 300 revolutions per minute, showing the speed every thirty seconds, L and L1 would be equal, but

if the indication was made once a minute L would be half the size of L1. Use 32-pitch gears here entirely.

L1 is a plate, not shown, that engages the inner end of pin L2. L2 projects through a circular slot in cam L3 and draws cam L3 along with it, L3 always following behind L2. L2 is carried on a crank L12, which revolves on the same axis as gear L1 and cam L3. Pin L2 carries a connecting rod L4, which carries a pivoted weight L5. As soon as crank L12 is a little past the upper dead-center, weight L5 throws it to the lower dead center. This throw should begin just before cam L3 engages the roller of L7, and during the throw the cam raises the roller and contact arm L7 and forces the lower iridio platinum contact against the upper one, closing the circuit, also breaking the contact just before the pin L2 comes to the lower dead center. So the contact lasts only for an instant, however slow the engine may be moving. This also makes it impossible for the contacts to be left together when the engine stops.

Adjusting the Engine Switch.—All the parts of this mechanism should work very freely. The weight should begin to fall before cam L3 has commenced to lift the roller, and during the fall the cam should go clear of the roller and break the contact at once.

Contact lever L7 should lift the lever L6 a full one-thirty-second of an inch, and when it breaks the contact they should separate the same distance. Pin L8 is a stop for arm L7, and pin L9 a stop for L6. Pin L10 is to prevent arm L7 from being lifted past L6. L7 should not quite strike pin L10 in its regular lift.

The springs of L7 and L6 should have very light pressure. If the weight L5 does not fall freely enough to make and break the contact instantly, see that the weight and connecting rod and cam L3 and crank L12 are all working perfectly free and that L7 does not strike pin L10; then see that L7 and L6 work so easy that it is impossible to leave the contacts together whenever the shaft of L may be stopped.

The Log Make-and-Break.—This is very similar to that of

the engine. The counter at the top is run by two pairs of worm gears. All the mechanism above the driving gear is mounted on a plate which is easily removed by taking the binding posts or nuts off the back. To oil the ball bearings remove the box cover, then the worm gears can be given a drop of oil also.

The counter in the log switch shows nautical miles to one-tenth of a mile. To get the log line to run steady a governor of the type supplied should be used. The governors in common use do not obtain steady enough run of the line for successful work with this indicator.

The Anemometer Make-and-Break.—This works on the same principle as the engine switch. Having motion only one way, cam L3 and crank L12 are made in one piece. What has been said about the free making and breaking of the contacts, distance of lift, and clearance in the engine switch holds good in the anemometer.

Packed oil cups are at the top and bottom of the cup shaft. The way to prevent squeaking is to keep it well oiled.

WHAT TO DO IF THE INDICATOR DOESN'T WORK RIGHT.

First. See that the battery is up to the required voltage—at least 8 volts for one indicator, and 16 for two.

Second. See that the wires and connections are free of breaks or short circuits.

Third. See that levers L6 and L7 are making good contact and getting a clear separation.

Fourth. If these are all right, then take off the record, and, without disturbing any adjustments, watch the working of each part of the indicator; and from the instructions already given, the difficulty should be easily located and the proper adjustment made.

Instruments of the above description were installed for test on the U. S. S. *Galveston* and were of great assistance in adjusting the engine speed for the various runs of her official standardization runs at Provincetown.

Sets of these instruments for the engines and also for the

log were temporarily installed on the U. S. S. *Maryland*. They are now being installed with certain improvements indicated by service on the vessel.

The instruments are found of very material assistance in quickly adjusting the engines to any desired number of revolutions and in indicating variations in the speed due to various causes, such as bad steering, heeling or drop in vacuum. If there is any change the pointer immediately indicates it, much more quickly and accurately than the eye and ear can do so.

SUGGESTIONS AS TO THE ADAPTABILITY OF A UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK.

(CONTINUED.)

By Luther D. Lovekin, Associate.

COMPOSITION FLANGES FOR PROPOSED UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK FOR PRESSURES 0 TO 1000 POUNDS, INCLUSIVE.

(See Drawings Nos. 1, 2, 3.)

To be used on all work where adjoining valves and fittings are of composition or other equally light proportions.

This is suggested so as to avoid the excess weight that would be required if the flanges were made to suit cast-iron valves and fittings. The latter, requiring a much larger diameter of boat circle and consequent outer diameter of flange, will be dealt with under a separate list in this article.

These flanges conform very closely to the ones used by the Bureau of Steam Engineering of the U. S. Navy, for all low-pressure work, and are equally suitable for the Bureau of Construction and Repair of the U. S. Navy, as well as merchant work, under these same conditions.

Owing to the slight variations that exist in the proportions of flanges, builders are required to have on hand various sizes of templets and patterns of flanges, as well as valves and other fittings, all of which involve considerable expense and annoyance to the builders in addition to the various departments in the U. S. Navy.

The object of this article is to bring about a discussion between the various parties interested, so that some step shall be taken to standardize the same.

The question of the strength of these flanges has not been dealt with on account of the very close proximation of this

suggested list, as previously stated, to the one used for several years by the Bureau of Steam Engineering, U. S. Navy.

This list is one which years of practice has shown to be of ample strength for the work required, while the weight of same is reduced to a minimum, and no further calculations seem necessary. The practical difficulties of attaching flanges to the pipes have to be considered fully when such light proportions are used, and from my experience in this line of work I do not believe we can do better than follow the facts of the case rather than enter into any new theory on these low-pressure flanges.

It is obvious that any malleable material might be used for these flanges, such as cast steel, wrought steel or iron, as well as malleable iron, with perfect satisfaction. When other than brass or bronze composition flanges are used, the brazing of the pipe to the flanges is not essential, provided the pipes are properly expanded into grooved flanges by approved machinery.

I append herewith a list of cast-iron flanges, which are intended for use where cast-iron valves and fittings are used, as previously mentioned.

CAST-IRON FLANGES FOR PROPOSED UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK FOR PRESSURES 0 TO 100 POUNDS.

(See Drawings Nos. 4, 5, 6, 7, 8, 9.)

This list is particularly well adapted for use where copper pipes are to be expanded into cast-iron flanges, on account of the sectional area of flange not being subjected to the same internal stresses as when wrought-iron or steel pipe is used; it saves weight, and should be encouraged by builders where possible.

For wrought-iron, lap-welded iron or steel pipe, or seamless-drawn steel pipe, it is advisable to combine the thicknesses H & L, and omit bosses on the back of flanges, so as to have more resisting power in the flanges; with sizes above 8 inches diameter, it is necessary, when expanding iron or steel pipe

into cast-iron flanges, to have the flanges turned on their outer diameter, so as to fit into clamping dies, to prevent flanges cracking when expanding. Hammer tests made from work done in this manner show the flanges to be practically free from any internal stresses; in fact, the metal in the flange seems to become more dense, and improve physically, when under the action of the powerful roller expander necessary to perform this work properly.

Numerous tests made by the author have shown that it is absolutely impossible to expand wrought-iron or steel pipe successfully into cast-iron flanges of the usual proportions without cracking same, unless the flanges are turned on the outside and fitted into dies which have their inner surfaces a counterpart of the outside of the flange; these dies are surrounded by a massive head on the expanding machine, which is greatly in excess of the compressive strength of the material in the pipe, thereby causing the metal in the pipe to flow into any grooves or recesses on the inside of the flange with perfect ease. This gives the flange great holding power, as will be seen from tests made by the U. S. Navy Department, and published in the JOURNAL, A. S. N. E., Vol. XVII, No. 2, 1905, and also "Marine Engineering Journal" of August, 1905.

It will be noted that the cast-iron flanges have been designed for tube dimensions, which vary considerably from pipe dimensions. The tube dimensions retain a constant diameter for the bore of the flange, while the pipe dimensions are determined by the inside diameter of the pipe. This list, however, serves for either tubes or pipes, and the small tables shown herewith give the corresponding pipe measurements that may be used. Thus, if we contemplate using a 12-inch pipe, we use the flange designed for a 12½-inch tube, etc., etc.

XU

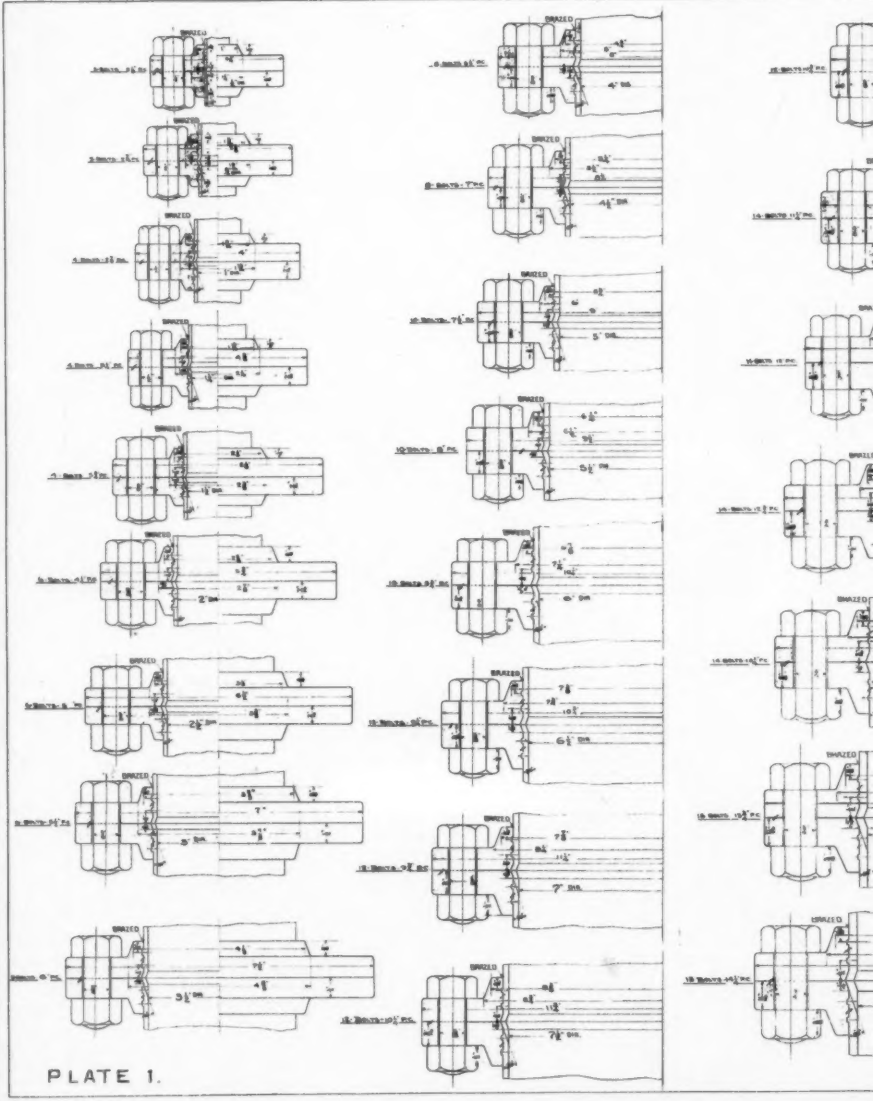
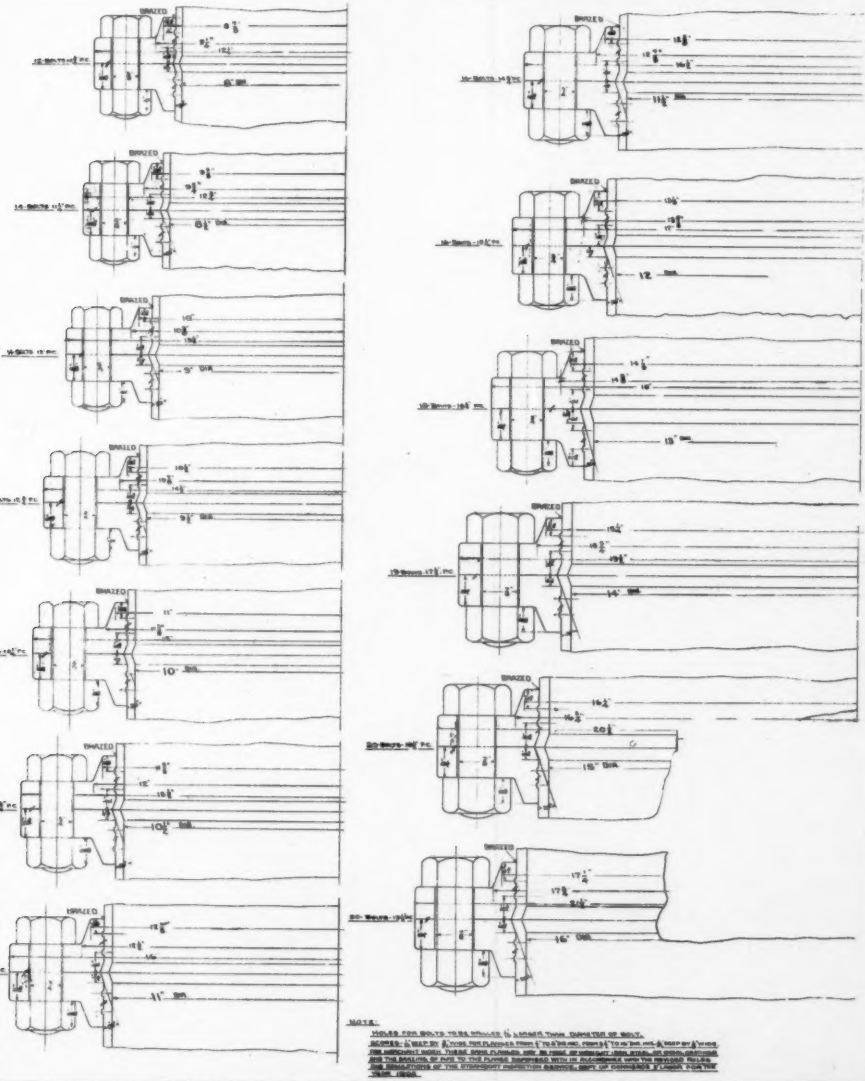


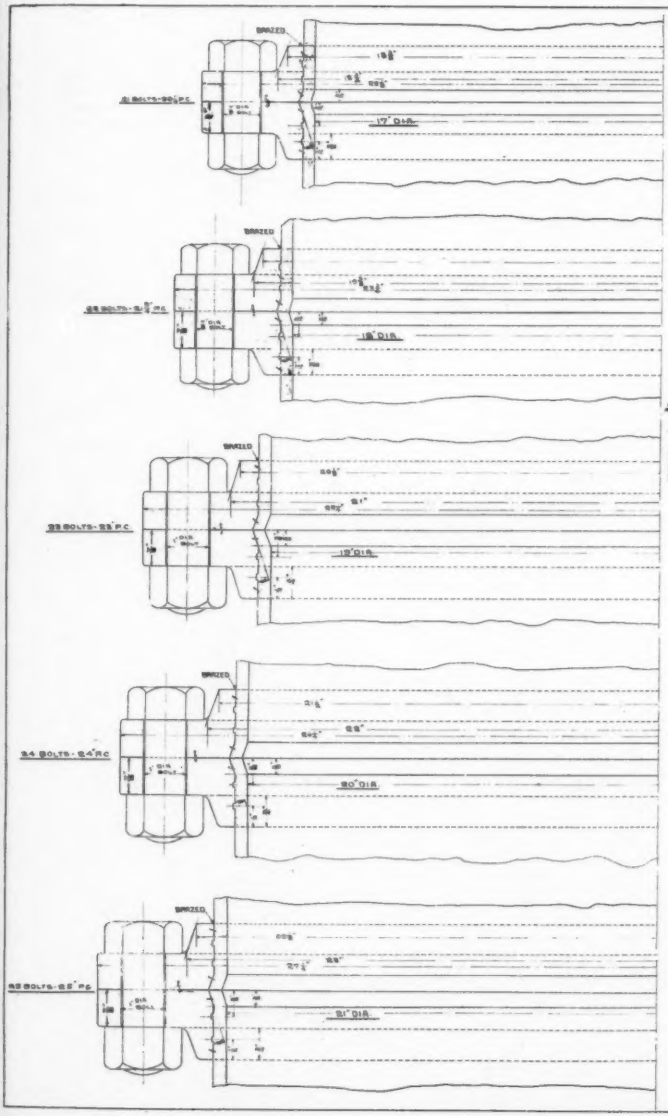
PLATE 1.



NOTE.

HOLES FOR BOLTS TO BE $\frac{1}{16}$" LARGER THAN DIAMETER OF BOLT.

FLANGES: $\frac{1}{2}$" THICK FOR FLANGES FROM 6" TO 8" INCL. FROM 8" TO 10" INCL. & $\frac{5}{8}$" THICK FOR 12" AND OVER. FOR HORIZONTAL MAINS, THESE SHOP FLANGES ARE TO BE USED AT HORIZONTAL IRON PIPES, AS SHOWN, AND THE BALANCE OF PIPE TO THE FLANGES SUPPLIED WITH IN ACCORDANCE WITH THE APPROVED RULES AND REGULATIONS OF THE UNDERWRITERS' INSPECTION BUREAU, DEPT. OF COMMERCE & LABOR, NEW YORK CITY, 1908.


BRACED
17" DIA.
BRACED
19" DIA.
BRACED
21" DIA.
BRACED
23" DIA.
BRACED
25" DIA.

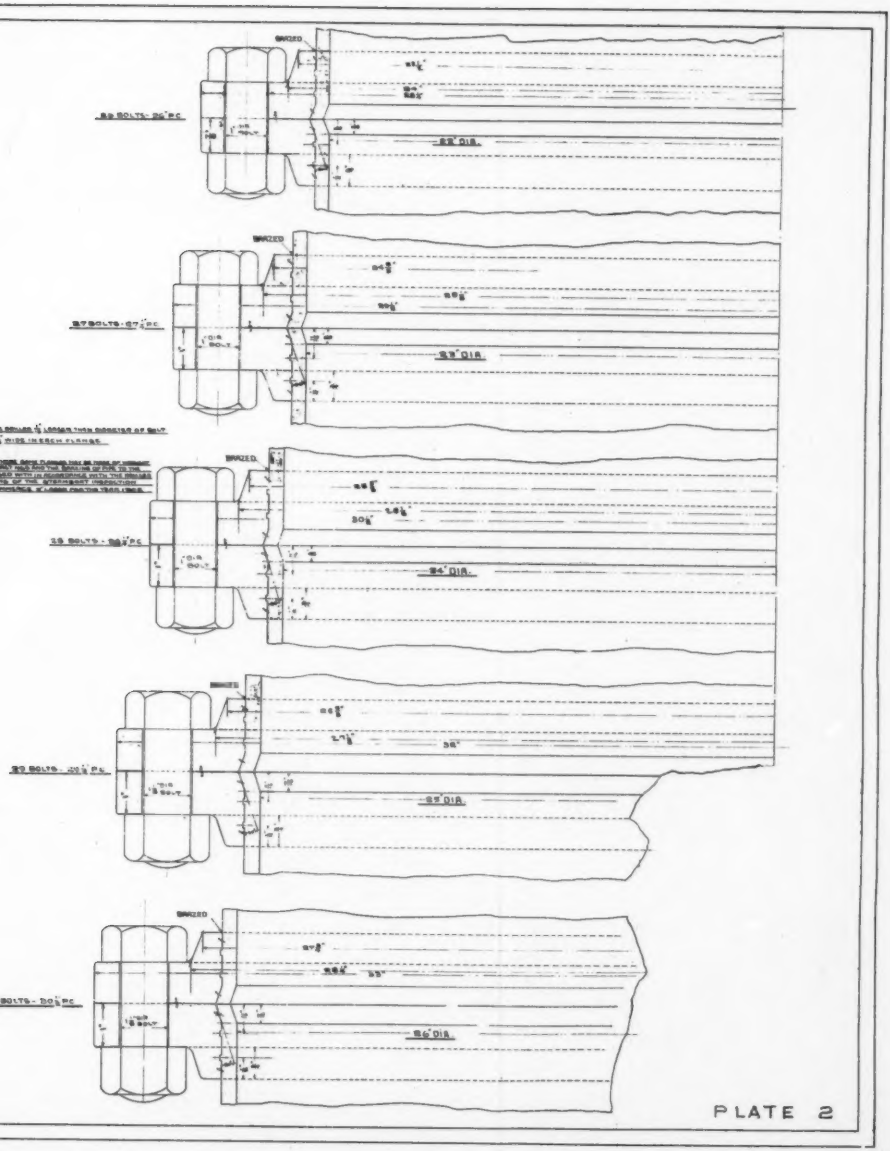

BRACED
22 DIA.
BRACED
23 DIA.
BRACED
24 DIA.
BRACED
25 DIA.
BRACED
26 DIA.

PLATE 2

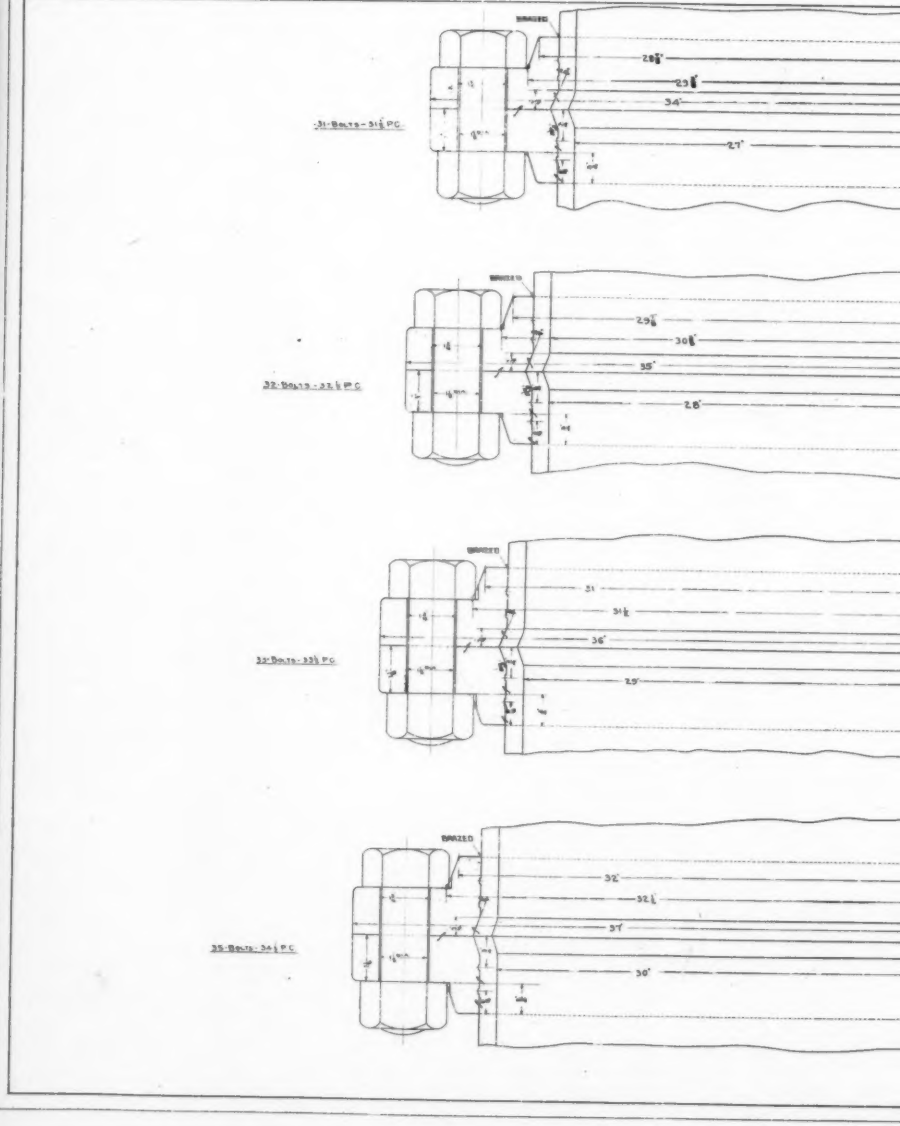
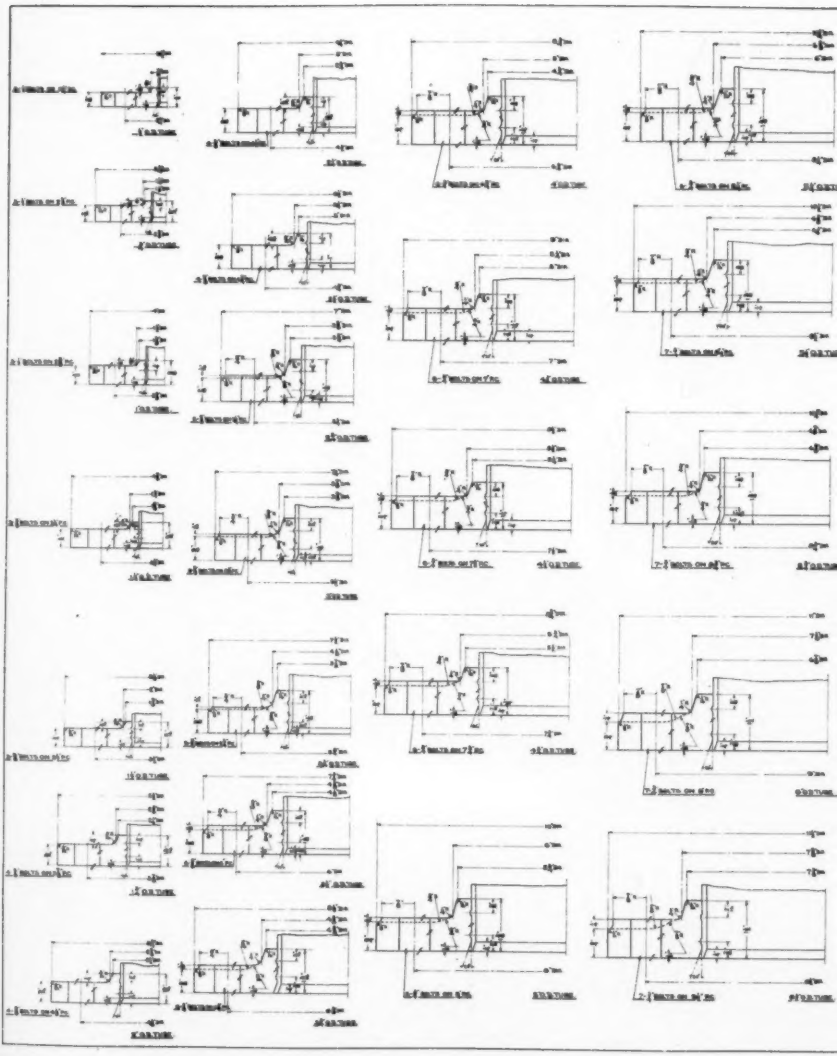

BRAZED
31-BOLTS - 31½ P.C.
29½"
29½"
34"
27"
BRAZED
32-BOLTS - 32½ P.C.
29½"
30½"
35"
28"
BRAZED
33-BOLTS - 33½ P.C.
31
31½
36"
29"
BRAZED
35-BOLTS - 34½ P.C.
32"
32½
31"
30"

PLATE 3

Note:
Two Scores ½ Deep & ¼ Wide in Each Flange.

FOR MERCHANT WORK THESE BRIDLE FLANGES MAY BE MADE OF WROUGHT IRON, STEEL, OR STEEL CASTINGS, AND THE BRAZING OF PIPES TO THE FLANGE DISPENSED WITH, IN ACCORDANCE WITH THE REVISED RULES AND REGULATIONS OF THE STEAMBOAT INSPECTION SERVICE, DEPT OF COMMERCE AND LABOR FOR THE YEAR 1908.



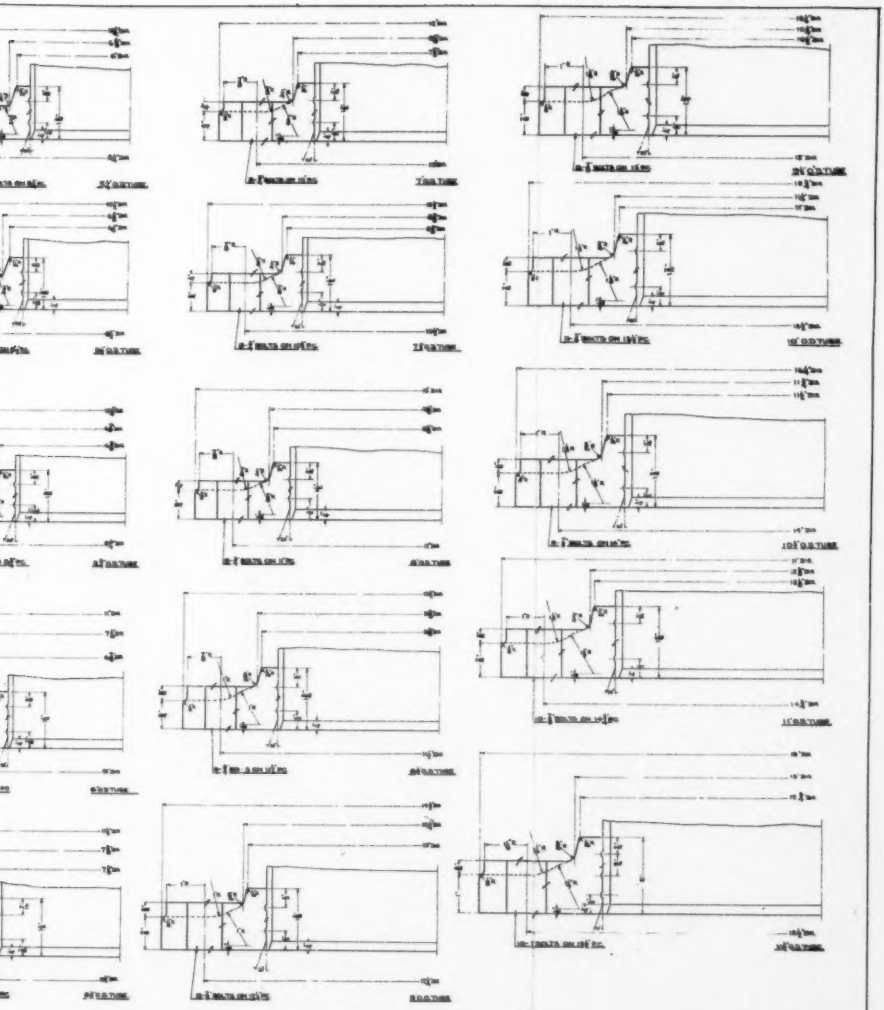
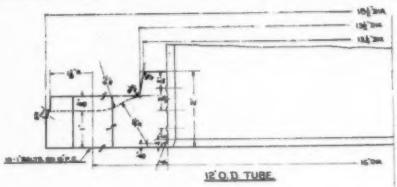
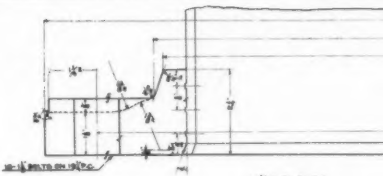


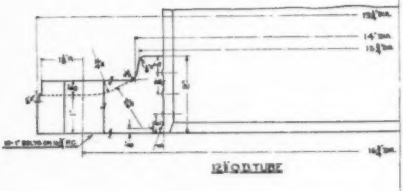
PLATE 4.



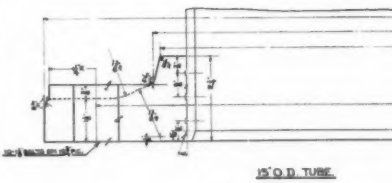
12" O.D. TUBE.



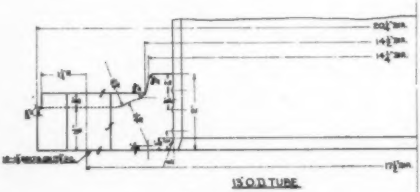
14" O.D. TUBE.



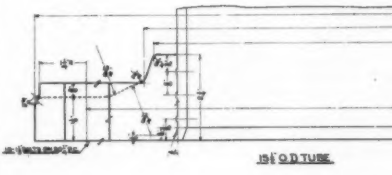
12½" O.D. TUBE.



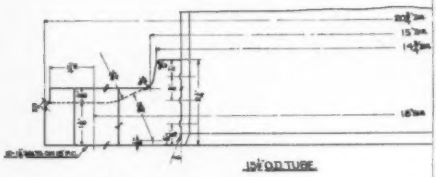
15" O.D. TUBE.



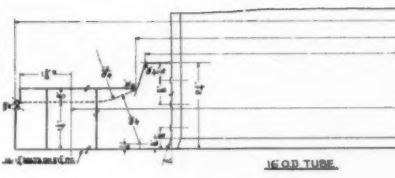
13" O.D. TUBE.



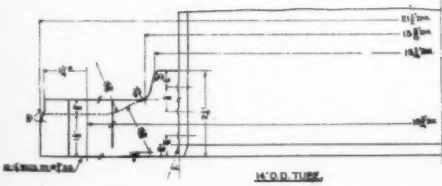
15½" O.D. TUBE.



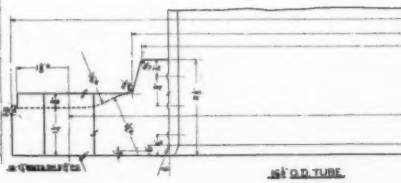
13½" O.D. TUBE.



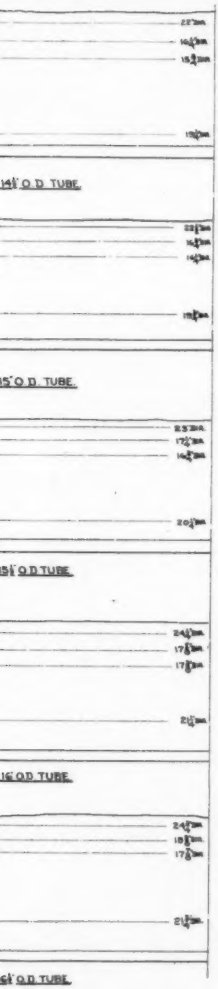
16" O.D. TUBE.



14" O.D. TUBE.



16½" O.D. TUBE.



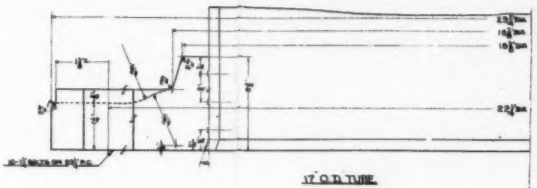
14½" O.D. TUBE.

15" O.D. TUBE.

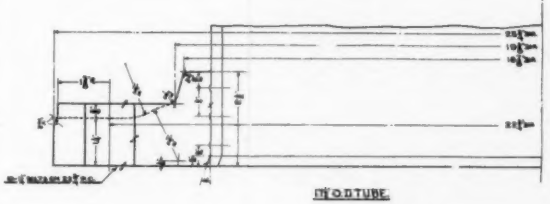
15½" O.D. TUBE.

16" O.D. TUBE.

16½" O.D. TUBE.

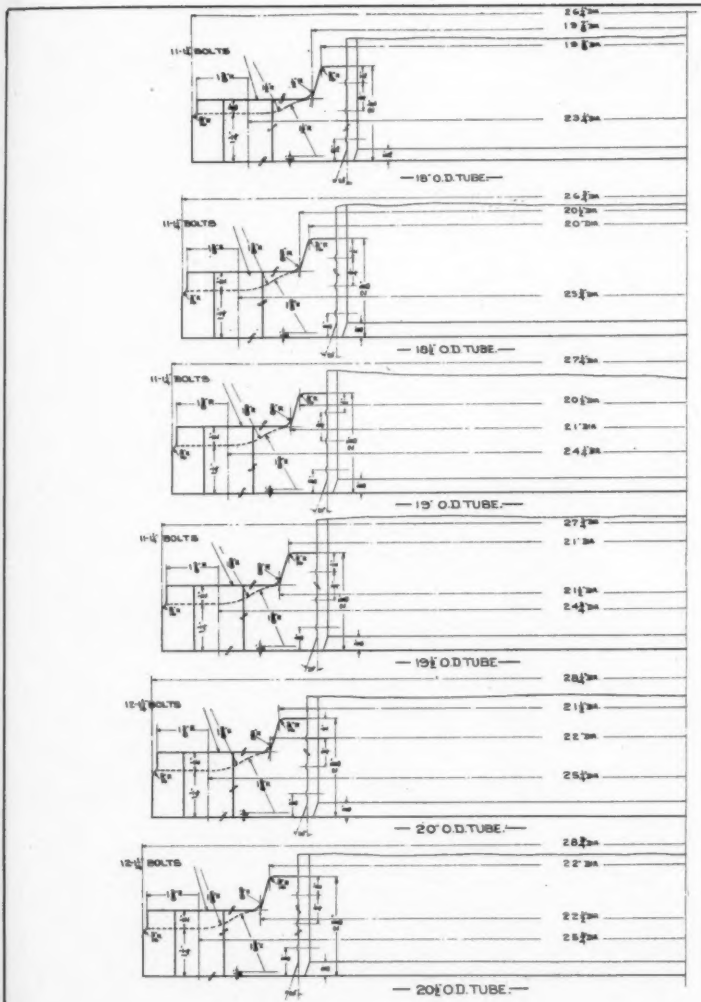


17" O.D. TUBE.



17½" O.D. TUBE.

PLATE 5


11-1¼" BOLTS
— 18" O.D. TUBE —
11-1¼" BOLTS
— 18½" O.D. TUBE —
11-1¼" BOLTS
— 19" O.D. TUBE —
11-1¼" BOLTS
— 19½" O.D. TUBE —
12-1¼" BOLTS
— 20" O.D. TUBE —
12-1¼" BOLTS
— 20½" O.D. TUBE —

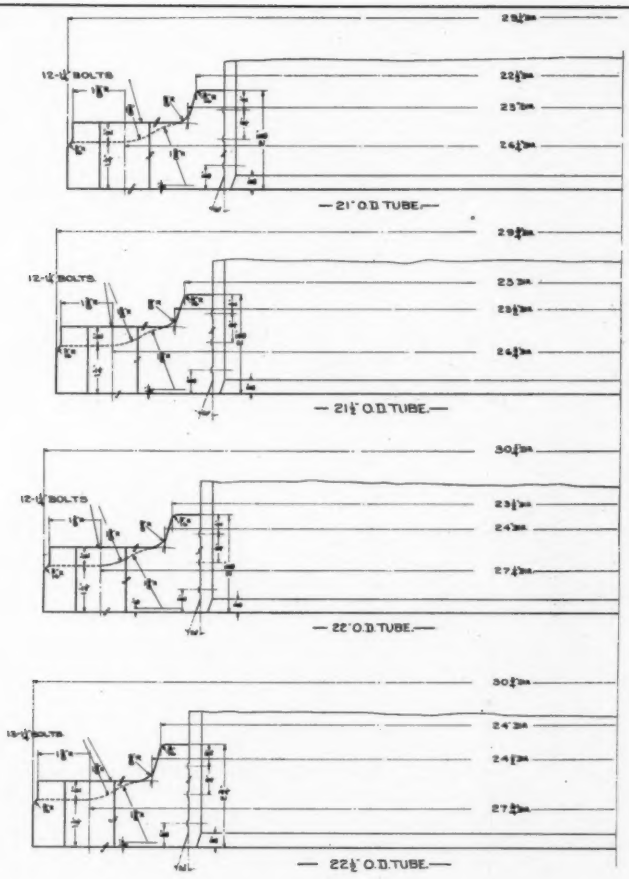
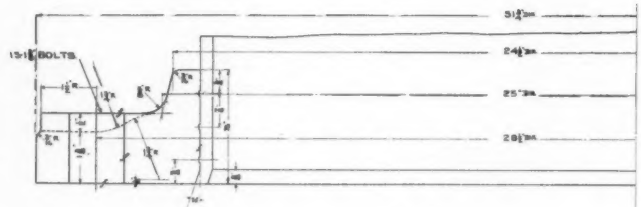
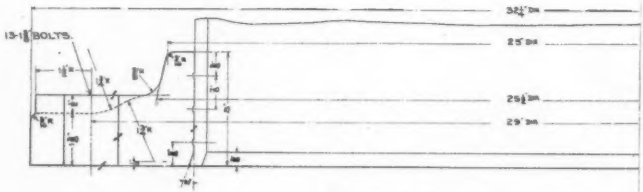

12-1" BOLTS
— 21" O.D. TUBE —
12-1" BOLTS
— 21½" O.D. TUBE —
12-1" BOLTS
— 22" O.D. TUBE —
12-1" BOLTS
— 22½" O.D. TUBE —

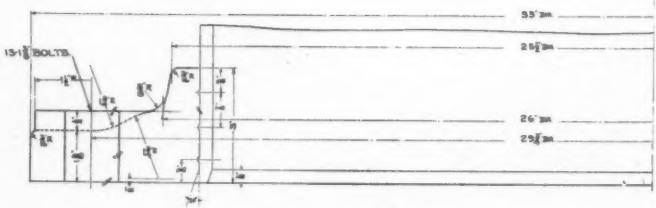
PLATE 6



—25" O.D. TUBE.—

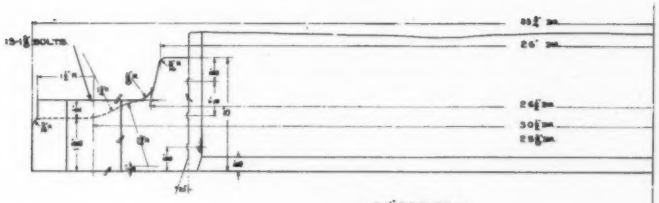


—25½" O.D. TUBE.—

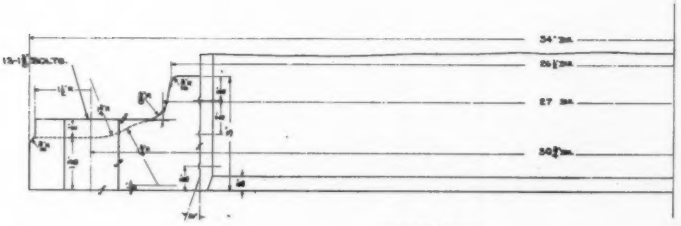


—24" O.D. TUBE.—

PLATE 7

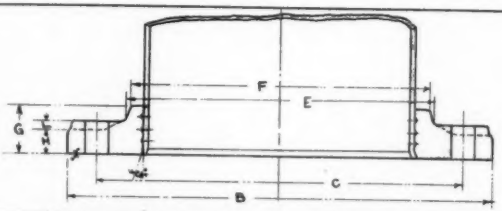


— 24" O.D. TUBE. —



— 25" O.D. TUBE. —

PLATE 8.

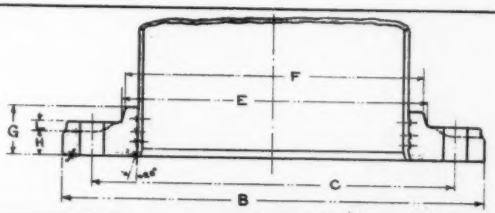


FROM 0 TO 100 LBS. PRESSURE PER SQ. INCH.

INSIDE DIA. OF PIPE	DIA. OF FLANGE B	DIA. OF BOLT CIRCLE C	E	F	G	H	L	DIA. OF BOLTS	NUMBER OF BOLTS	M	N	O	P	
¼	3½	2¾	7/8	7/8	½	⅝	—	½	3					
⅜	3½	2⅞	1½	1⅛	9/16	11/16	—	½	3					
½	4	2⅞	1½	1⅛	⅝	½	—	½	3					
1	4⅞	3¼	1¾	1⅜	11/16	½	—	⅝	3					
1¼	5⅛	3½	2	1⅝	11/16	½	—	⅝	3					
1½	5½	3⅞	2⅛	2¼	13/16	9/16	—	⅝	4					
1¾	5¾	4⅛	2⅜	2½	13/16	9/16	—	⅝	4					
2	6½	4½	3	2¾	15/16	⅝	—	¾	4					
2¼	6½	4¾	3¼	3	1	⅝	—	¾	4					
2½	7	5¼	3⅝	3⅜	1⅛	⅝	15/16	¾	5					
2¾	7½	5½	3⅞	3⅝	1⅛	⅝	15/16	¾	5					
3	7½	5¾	4⅛	3⅝	1⅛	⅝	15/16	¾	5					
3¼	7¾	6	4⅜	4⅛	1⅛	⅝	⅞	¾	6					
3½	8¼	6¼	4⅝	4⅜	1 1/16	⅝	⅞	¾	6					
3¾	8½	6½	5	4½	1¼	¾	⅞	¾	6					
4	9	7	5¼	5	1¼	¾	⅞	¾	6					
4¼	9½	7½	5½	5¼	1¼	¾	1	¾	6					
4½	9¾	7¾	5¾	5½	1¼	¾	1	¾	6					
4¾	10	8	6	5¾	1⅜	¾	1	¾	6					
5	10¼	8¼	6⅛	6	1⅜	¾	1	¾	6					
5¼	10½	8½	6⅜	6¼	1⅜	¾	1	¾	7					
5½	10¾	8¾	6⅝	6½	1⅜	¾	1	¾	7					
5¾	11	9	7⅛	6¾	1½	¾	1¼	¾	7					
6	11½	9½	7⅜	7⅛	1½	¾	1¼	¾	7					
6½	12	10	8⅛	7⅝	1½	¾	1¼	¾	8					
7	12½	10½	8⅜	8⅛	1½	¾	1¼	¾	8					
7½	13	11	9⅛	8⅝	1½	¾	1¼	¾	8					
8	13½	11½	9⅝	9⅛	1⅝	¾	1⅜	¾	9					
8½	14½	12½	10¼	10	1⅝	⅞	1⅜	⅞	9					
9	15¼	13	10¾	10½	1⅝	⅞	1⅜	⅞	9					
9½	15¾	13½	11¼	11	1⅝	⅞	1⅜	⅞	9					
10	16¼	14	11¾	11½	1⅝	⅞	1⅜	⅞	9					
10½	17	14¾	12⅜	12⅛	1⅝	⅞	1⅜	⅞	10					

INSIDE DIA. OF PIPE	DIA. OF FLANG B
11	18
11½	18½
12	19
12½	20
13	20½
13½	21½
14	22
14½	22½
15	23
15½	24
16	24½
16½	25½
17	25½
17½	26½
18	26½
18½	27½
19	27½
19½	28½
20	28½
20½	29½
21	29½
21½	30½
22	30½
22½	31½
23	32½
23½	33
24	33½
24½	34

PLATE 9.



FROM 0 TO 100 LBS. PRESSURE PER SQ. INCH.

INSIDE DIA. OF PIPE	DIA. OF FLANGE B	DIA OF BOLT CIRCLE C	E	F	G	H	L	DIA OF BOLTS	NUMBER OF BOLTS	M	N	O	P	
11	18	15½	13	12¾	2	1	⅜	1	10					
11½	18½	16	13½	13¼	2	1	⅜	1	10					
12	19½	16¾	14	13½	2	1	⅜	1	10					
12½	20¼	17½	14½	14¼	2	1⅛	⅜	1⅛	10					
13	20½	18	15	14½	2¼	1⅛	⅜	1⅛	10					
13½	21½	18¾	15⅜	15¼	2¼	1⅛	⅜	1⅛	10					
14	22	19½	16¼	15½	2¼	1⅛	⅜	1⅛	10					
14½	22½	19¾	16¾	16¼	2¼	1⅛	⅜	1⅛	10					
15	23	20¼	17¼	16¾	2¼	1⅛	⅜	1⅛	10					
15½	24¼	21¼	17⅞	17¼	2¼	1¼	⅜	1¼	10					
16	24¾	21¾	18⅜	17¾	2¼	1¼	⅜	1¼	10					
16½	25¼	22¼	18⅞	18¼	2¼	1¼	⅜	1¼	10					
17	25¾	22¾	19⅜	18¾	2¼	1¼	⅜	1¼	10					
17½	26¼	23¼	19⅞	19¼	2¼	1¼	⅜	1¼	11					
18	26¾	23¾	20¼	20	2⅜	1¼	½	1¼	11					
18½	27¼	24¼	21	20½	2⅜	1¼	½	1¼	11					
19	27¾	24¾	21½	21	2⅜	1¼	½	1¼	11					
19½	28¼	25¼	22	21½	2⅜	1¼	½	1¼	12					
20	28¾	25¾	22½	22	2⅜	1¼	½	1¼	12					
20½	29¼	26¼	23	22½	2⅜	1¼	½	1¼	12					
21	29¾	26¾	23½	23	2⅜	1¼	½	1¼	12					
21½	30¼	27¼	24	23½	2⅜	1¼	½	1¼	12					
22	30¾	27¾	24½	24	2⅜	1¼	½	1¼	13					
22½	31¾	28¼	25	24½	3	1⅜	½	1⅜	13					
23	32¼	29	25½	25	3	1⅜	½	1⅜	13					
23½	33	29¾	26	25½	3	1⅜	½	1⅜	13					
24	33½	30¼	26½	26	3	1⅜	½	1⅜	13					
24½	34	30¾	27	26½	3	1⅜	½	1⅜	13					

SOME STERN FACTS.

By Horace See, Associate.

(Paper read at the Milan meeting of the International Navigation Congress.)

It is remarkable that the natural laws are so often disregarded, particularly in the mechanic arts, and in no branch is it more strikingly illustrated than in the vessel driven by the screw propeller. We might say that here fantastic shapes are the most popular, as shown in the propeller and the stern of a vessel. The propeller, like the Indian's boat, has received a dig from every passer-by, and all sorts of shapes suggested for obtaining the highest efficiency. The expanding pitch with the tip of blade bent aft radially so as to assume a form like a spoon was at one time a prime favorite, but today has given place to the true screw. The latter, after a thorough investigation, was accepted by the writer to be the one, when properly designed, best suited for propelling a vessel, whether the stern was full or lean, as it offered the least resistance in its path whilst obtaining maximum effect. The success with the bronze blade is in the same line, as it is to the thin blade offering less resistance than the thicker iron one, and not to the kind of metal, that this result is due. The frequency, therefore, with which this partiality for the spoon-shaped blade was expressed led the writer, when a contract had been obtained some twenty odd years ago for two vessels similar in every respect, to suggest, in order to solve the problem, that one be fitted with a blade in the form of a true screw and the other with one spoon shaped. These wheels were made solid, the outline as well as thickness of the blades being the same, the only difference being that the one was bent to form a true screw, whilst the other was expanded or twisted to make the spoon shape.

When the vessels were ready to launch with the wheels in place they were visited by quite a number of persons, from whom the general expression was in favor of the spoon-shaped blade, as it was said by them to take hold of the water better than the true screw. This view was quite true in one way, but not in that desired for obtaining the best results in propelling the vessel, as the one with the true screw was driven at a speed one knot greater than that with the spoon-shaped wheel. This result could not be attributed to the form of the one vessel or the character of its machinery being superior to that of the other, or to the handling of the vessels, but to the wheels themselves, as a new wheel of the true-screw pattern when it replaced the spoon-shaped one brought the speed of this vessel up to that of the other. Again, when the one first fitted with the true screw broke this wheel and had the spoon-shaped one that had been taken off the other vessel placed on it, then her speed also was reduced as in the case of the vessel originally fitted with this wheel.

A tunnel arched longitudinally with its lowest point at bottom of vessel forward of the screw, increasing and attaining its highest point over it and then descending to a point at or below the water line at the stern, is another form that has fascinated many. The United States Government has had a number of vessels of this type built, the contractors for which have lost a considerable sum of money—some having gone into bankruptcy—by failing to secure the contract speed. The original and altered form of stern are shown in Fig. 1. These boats were designed to maintain a speed of 29 knots for two consecutive hours. The *Perry* having attained but 28.2 knots on the progressive speed trials, the contract speed demanded was afterwards reduced to 26 knots, and the duration of run to one hour. The hook at stern was also removed and new propellers placed on two of the vessels. The *Paul Jones* gave the best general results—attaining after the change 28.91 knots on the progressive speed trials and 27.4 on the one-hour's run. The location of propellers and the form of run, still being bad, worked against attaining a higher

FIG. I.

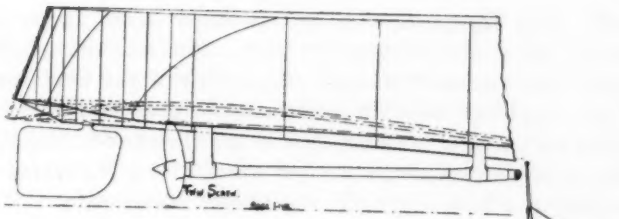


FIG. II.

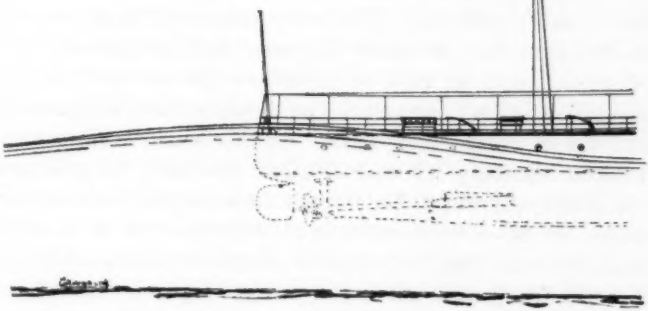


FIG. III

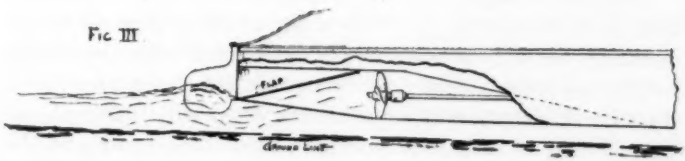
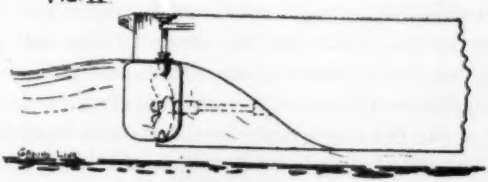


FIG. IV



speed. The change of trim in this vessel is not given, but that of the others averaged 5½ feet. The selection of this form of stern is more remarkable in view of the experience in 1889 with the dynamite-gun cruiser *Vesuvius*, contracted to attain a speed of 20 knots per hour, when tried in 4½ to 5 fathoms or comparatively shallow water. The trials, four in number, with this depth of water were unsuccessful. The writer, who was present only on the last one, suggested after that failure a course where the depth of water would not be less than 15 fathoms. This met with favor, and when the next or final trial was made the vessel readily attained a speed of 21.65 knots or 1.65 in excess of the contract requirements. She had extremely fine lines, with a wedge-shaped and not an overhanging stern. On this occasion she stood up without squatting aft and with but very little disturbance in the water, whereas, in the former trials the stern not only was depressed, but a wave was formed with the crest of it so high as to cover the deck at this point, as shown in Fig. II, thereby demonstrating that the water where depth is limited will, when leaving the stern, ascend naturally without mechanical means.

This now brings me to the tunnel stern with an adjustable flap, as shown in Fig. 3. Why a flap at all, is difficult to understand, not only in view of the experience with the above-mentioned torpedo-boat destroyers, but also from the experiments with a shallow-draught launch fitted with a tunnel and hinged flap, of which it is stated in the Transactions of the Institution of Naval Architects, of 1903, page 109: "When the launch is light, drawing 11 inches, with the same power the speed is increased from 9.2 miles an hour with the flap down to 10 miles an hour with the flap up; and when the draught is 28 inches, loaded with 20 tons, the speed is increased from 6.9 miles an hour with the flap down to 8.25 miles an hour with flap up, the power at both speeds being the same. As might naturally be expected, the increase of efficiency due to the lifting of the fan is greater when the boat is loaded, the lower speed in each case being what it would have been if

there had been no adjustable flap, clearly showing the advantage of the flap." Is not this an admission that the vessel would be better without the flap, meaning the same as the expression that "a dead Indian is the best Indian?" Then why should there be a flap if the boat does better with it raised? Why should it be employed to make the tunnel longer, thereby decreasing the displacement and with it increasing skin resistance? It is also said "that the rush of water being in an inclined direction towards the bottom, rather tends to scour it," but there is a failure to inform us whether the scouring leaves a true or a roughened surface. It is perfectly natural to suppose the latter is the case, and that the flap, not only a disadvantage but also an unnecessary appendage, did project the water downwards to form bridge holes, thereby retarding the vessel in shallow water and demanding greater power to do the work, a result similar to what it would be in the case of a vessel with a roughened bottom, as described in the paper of Capt. Suppan before the Seventh Congress. A screw, however, working in a tunnel with the air from the sides excluded and a free discharge aft seems to be a more desirable formation. It also seems that if the after portion of each side is made to form a rudder, as shown in Fig. IV, an additional advantage is gained in being able to steer a straighter course, as the full surface of the rudders is brought into play, a result not accomplished with the flap, as such a vessel is said to steer badly, the reason, no doubt, being that the water is projected downwards and away from the surface of rudder in proportion to the depression of the flap. With double or twin rudder its position in this arrangement not only affords protection, but also permits the quadrant or tiller being placed below the deck and also in a protected position.

The conclusions drawn by me from the above *in re* the movement of a vessel through a fluid have been as follows:

1. That all vessels create one or more waves.

2. That their character at the stern is partly molded by the shape at this point.

3. That the height of waves above the normal water level

is governed by the speed of vessel through the water and the latter's depth under the vessel.

4. That the wave at stern in shoal water answers the purpose of a flap, without restricting the flow aft and with least resistance.

5. That the discharge from a screw propeller should not only be free, but without restriction aft of it, no matter what the depth of water or at what speed the vessel moves through it.

6. That discharging the water downward at the stern will scour the bottom and form bridge holes to a greater extent than if discharged directly astern.

7. That the depth of bridge holes will be in proportion to the intensity of the discharge and proximity of the wheel to the bottom.

8. That a tunnel or form of stern directing the water to the screw propeller is valuable when its run limits the amount of skin friction and when it expands aft of the propeller.

9. That the propeller should be placed sufficiently far aft to limit the skin friction from the stream set in motion.

U. S. BATTLESHIP *VIRGINIA.*

By Lieutenant C. A. Gardiner, U. S. Navy, Member.

The *Virginia* is one of the three battleships authorized by Act of Congress approved March 3, 1899, the others being the *Nebraska*, building at Moran Bros., Seattle, Wash., and the *Georgia*, building at the Bath Iron Works, Bath, Me.

The *Virginia* was built by the Newport News Shipbuilding and Dry Dock Company under contract signed February 15, 1901. This contract provided that the ship should be delivered February 15, 1904, but the time was extended to October 31, 1905.

The total cost of the vessel, excluding armament, certain portions of the armor, etc., is $3,590,000, of which $1,240,000 is for the machinery.

The guaranteed speed is 19 knots per hour, maintained for four consecutive hours, on a mean draught of 23 feet 9 inches, corresponding to a displacement of about 15,000 tons. The usual penalties for failure to attain this speed are attached.

The keel was laid May 21, 1902, and the vessel launched April 5, 1904. The hull is made of basic open-hearth steel, with the frames spaced four feet apart, except under the machinery compartments. The inner bottom extends from frame 13 to frame 95. The double-bottom compartments between frames 60 and 69 are arranged for reserve fresh-water tanks. A coffer dam, 36 inches wide, extends the whole length of the ship on the protective deck, extending 3 feet above the berth deck, packed with corn-pith cellulose forward and abaft the midship armor. All woodwork, except keels, armor backing, etc., is fireproofed. All unexposed decks are covered with linoleum.

The superstructure extends between the two 12-inch turrets, but is not the full width of the main deck, leaving a wide

gangway on each side. It is recessed on each side just forward of amidships to permit training of the 8-inch turrets, and cut away at the corners to give greater train to the 3-pounder guns on the bridge deck.

BRIDGE DECK.

The bridge deck forms the upper part of the superstructure. Over the forward end is built the forward bridge, on which are the chart house, the emergency cabin, and the entrance to the conning tower. Two 3-pounder, two 30-caliber automatic, and two 30-caliber machine guns are mounted on this bridge. Over this is the flying bridge, with steering and standard compasses and two searchlights. Over the after end of the bridge deck is the after bridge, on which are mounted two 3-pounder and four 30-caliber automatic guns. The following boats are carried on the bridge deck:

Forward on starboard side and on port side, one 36-foot launch, one 30-foot cutter, one 20-foot dinghy, stowed one within the other, and each nest hoisted out by a 5-ton electric crane. Amidships, on starboard side, one 50-foot steam cutter, and aft, on the starboard side, one 16-foot dinghy inside one 30-foot cutter, all three hoisted out by a 15-ton crane. Amidships, on port side, one 36-foot steam cutter, and aft, on port side, one 30-foot cutter, both hoisted out by an 8½-ton crane. Four 3-pounders are mounted forward on this deck, and the same number aft.

MAIN DECK.

This deck extends the full length of the vessel, the superstructure covering it from about frame 32 to frame 78. The two 12-inch turrets with 8-inch superposed turrets, and the two 8-inch amidships turrets, are on this deck, and in the superstructure there are mounted four 3-inch R.F. guns. The hawse-pipes enter this deck through the bow, the anchor chains leading to the windlass house, which projects above this deck. A billboard and an anchor crane for each of the sheet anchors are on this deck. The bower anchors are of the Dunn patent variety, and stow in the hawse-pipes. Two 30-

foot cutters and two 20-foot dinghies are slung to davits forward, and two 30-foot whaleboats, a 30-foot barge and a 30-foot gig aft. The following are also found on this deck in order from forward: crew's galley, drying room, bakery, officers' galley, paymaster's office, executive officer's office, officers' bath rooms, officers' water closets, wireless telegraph station.

GUN DECK.

On this deck is mounted the 6-inch battery of twelve guns, also four 3-inch R.F. guns forward, and the same number aft. Forward are the crew's shower baths and lavatories, amidships the berthing spaces, and aft the quarters of the Admiral, Captain, Chief of Staff, Executive, Navigator, Chief Engineer, Paymaster, and two wardroom rooms.

BERTH DECK.

On this deck, besides the crew space amidships, there are forward, the sick bay, operating room, isolation ward, dispensary and bath, chief machinists' and chief petty officers' quarters and lavatories, master-at-arms' and sergeant of marines' staterooms, laundry, refrigerating plant and brigs, and aft, the officers' quarters.

PROTECTIVE DECK.

Extends the full length of the ship, and is 3 inches thick on the slope and 1½ inches on the flat. Above it is the cofferdam previously mentioned.

LOWER DECKS.

On the upper platform deck are the magazines, store rooms, sail room, etc., except the 12-inch magazines and shell rooms and forward and after 6-inch shell rooms, which are on the lower platform. In the hold are 6-inch magazines, mine room, etc.

ARMOR.

The side belt is a complete water-line belt 8 feet in width. This belt is of maximum thickness for a distance of 192 feet

abreast the engines and boilers, 11 inches thick at its upper edge, and maintaining this thickness downward for a distance of 5 feet, from which point it tapers to 8 inches thick at its lower edge. This belt is completed to the bow and stern as follows: for a distance of 60 feet forward and 32 feet abaft the heavy belt above described, the plates are 9 inches in thickness at the upper edge, tapering uniformly to 6 inches in thickness at the lower edge. The next course of plates forward and aft, for a distance of 16½ feet, are 6 inches in thickness at the upper edge, tapering uniformly to 4½ inches at the lower edge. The next course of plates forward and aft, for a distance of 17 feet, are 5 inches at the upper edge, tapering to 4 inches at the lower edge. The belt is completed to the bow and stern with plates of a uniform thickness of 4 inches.

Above the main belt, and covering the central portion of the hull for a distance of 245 feet, the ship's sides are protected by armor of a uniform thickness of 6 inches, extending up to the main deck, this upper and lower-casemate side armor being joined to the barbette of the after 12-inch turret by athwartship armor 6 inches thick, and connected forward by inclined armor 6 inches thick, thus forming a central casemate.

The conning tower is 9 inches in thickness, size, in the clear, 7 by 11 feet, and fitted complete, with space for steering wheel, rudder and shaft indicator, engine-room and other telegraphs, telephones, electric bells, speaking tubes, etc. The signal tower is 5 inches in thickness and 6 feet diameter in the clear.

ARMAMENT.

The 12-inch guns are mounted in pairs in two elliptical, electrically-controlled, balanced turrets, each pair of guns having an arc of fire of 270 degrees. The 8-inch guns are mounted in pairs in four elliptical turrets. Two of these are superposed on the 12-inch turrets, forming part thereof. The other two 8-inch turrets are balanced, electrically-controlled, and placed one on each beam; each pair of guns so mounted having an arc of fire of 180 degrees. The 12-inch turret

armor is as follows: 12 inches on the slope, 8 inches on the side, 10 inches on the front, and 6 inches on the back. The 8-inch turret armor is 12 inches on the slope, 8 inches on the side, 10 inches on the front, 6 inches on the back, and 2 inches on the top. The 6-inch guns are mounted singly on pedestal mounts, with shields in a central armored casemate on the gun deck, each gun having an arc of fire of 110 degrees. The secondary battery consists of twelve 3-inch R.F. guns, twelve 3-pounders, semi-automatic guns, four 1-pounder heavy automatic guns, four 1-pounder heavy R.F. guns, two 3-inch field pieces, two .30-caliber machine guns, six .30-caliber automatic guns, and four submerged torpedo tubes, two forward and two aft.

Two Barr and Stroud mechanical range-finders are mounted, one in each lower top.

HULL DATA.

Length between perpendiculars, feet	435
on load-water line, feet	435
over all, feet and inches	441–3
Beam, extreme, feet and inches	76–2½
at load-water line, feet and inches	76–2½
molded, feet	76
Ratio of length to beam	5.708
Depth, molded, main-deck side at M.S., feet and inches	41–3½
Draught, normal, mean, feet and inches	23–9
Displacement, mean load, tons	14,979
full load, tons	16,094
per inch at L.W.L., tons	61.1
Area of midship section to L.W.L., square feet	1,725
L.W.L. plane, square feet	23,650
Wetted surface, square feet	39,200
Center of gravity of L.W.L. plane aft of M.S., feet and inches	3–3½
above bottom of keel, feet	26.25
buoyancy above bottom of keel, feet	13
forward of midships, feet	105
Transverse metacenter above center of buoyancy, feet	19.18
Longitudinal metacenter above center of buoyancy, feet	519
Number of frames	108
Coefficient of fineness, block	.6696
M.S.	.9618
L.W.L.	.7724

MAIN ENGINES.

There are two four-cylinder, triple-expansion, inward-turning engines of the vertical, inverted, direct-acting type, in two watertight compartments, separated by a fore-and-aft bulkhead. The order of the cylinders is F.L.P., H.P., I.P., A.L.P. The H.P. and I.P. cranks are at 90 degrees, as are also the two L.P. cranks, and the sequence of cranks is H.P., I.P., F.L.P., A.L.P. The cylinders for each engine are supported by six front and six back columns, flanged and bolted at the bottom to the bed plate, and at the top to the cylinder casings. For each athwartships pair of columns there are two symmetrically arranged diagonal braces, secured at the ends to the columns, and at their centers to each other, and also a horizontal tie rod with sleeve distance piece. The columns are strengthened by horizontal and diagonal fore-and-aft tie rods. All the above of forged steel. The H.P. and I.P. cylinders are so connected as to prevent athwartships motion while permitting fore-and-aft play.

The valves are the single-ported piston type, with cast-iron heads and packing rings and steel followers. The H.P. is an inside valve, the others, outside. The H.P. cylinder has one valve, the others, two each, all arranged to lift out at the top of the valve chest. Each valve has a balance piston working in a cylinder cast with the upper valve-chest cover. Stephenson double-bar link valve gear is fitted, and the I.P. and L.P. valve stems are connected to crossheads.

The floating-lever steam reversing engine with oil-control cylinder is used, and each link is fitted with an independent linking-up gear of the usual pattern, giving a range of cut off of from .48 to .74 the stroke. The valve data is given elsewhere.

Reversing Engine.—Diameter steam cylinder, inches, 16; diameter oil cylinder, inches, 8; stroke, 22 inches.

The cylinders and valve chests are of hard cast iron with cast-iron linings. A jacket space of three quarters of an inch is left between the cylinder casings and their linings. Steam at working pressure enters the H.P. jacket, from which it

passes through reducing valves to the I.P. and L.P. jackets, each jacket being fitted with a drain pipe leading to an automatic trap.

The H.P. piston is of cast iron, the I.P. and L.P. pistons of cast steel, all having forged-steel followers. The H.P. and I.P. pistons have one cast-iron packing ring, which is kept expanded against the cylinder by springs operating longitudinally on the ends of the ring. The L.P. packing rings are two in number for each piston, and each ring is in eight segments set out by steel springs. The rings break joints and are secured to each other by dowel pins.

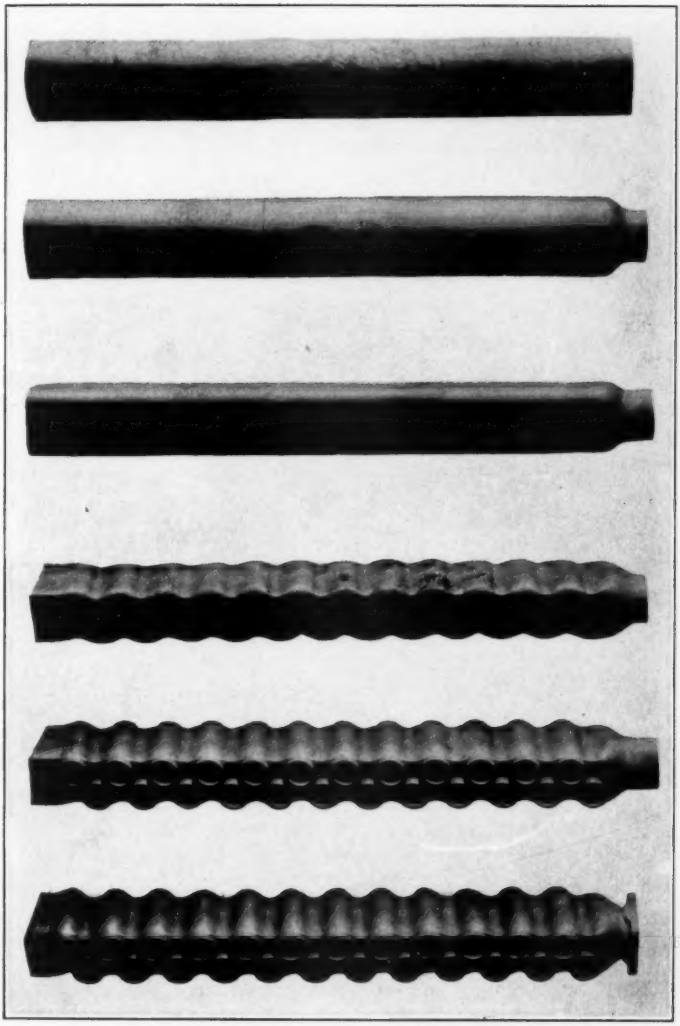
The piston rods are of forged steel, hollow, accurately ground and polished, and tapered at each end to fit the piston and the crosshead. The crossheads are of steel, forged with the crosshead pins, and are bolted to the cast-steel slippers. The crosshead guides, both ahead and astern, are of cast steel. The latter are bolted to flanges on the former, and these guides are bolted to brackets on the cylinder casings. The crank-pin brasses are of Navy standard composition, lined with white-metal, the brasses and steel caps secured by through bolts to the connecting-rod ends.

The eccentrics are made in two parts, the larger of cast-iron, the smaller of forged steel. The eccentric straps are of Navy standard composition, lined with white metal, and are secured to the forged-steel eccentric rod by stud bolts.

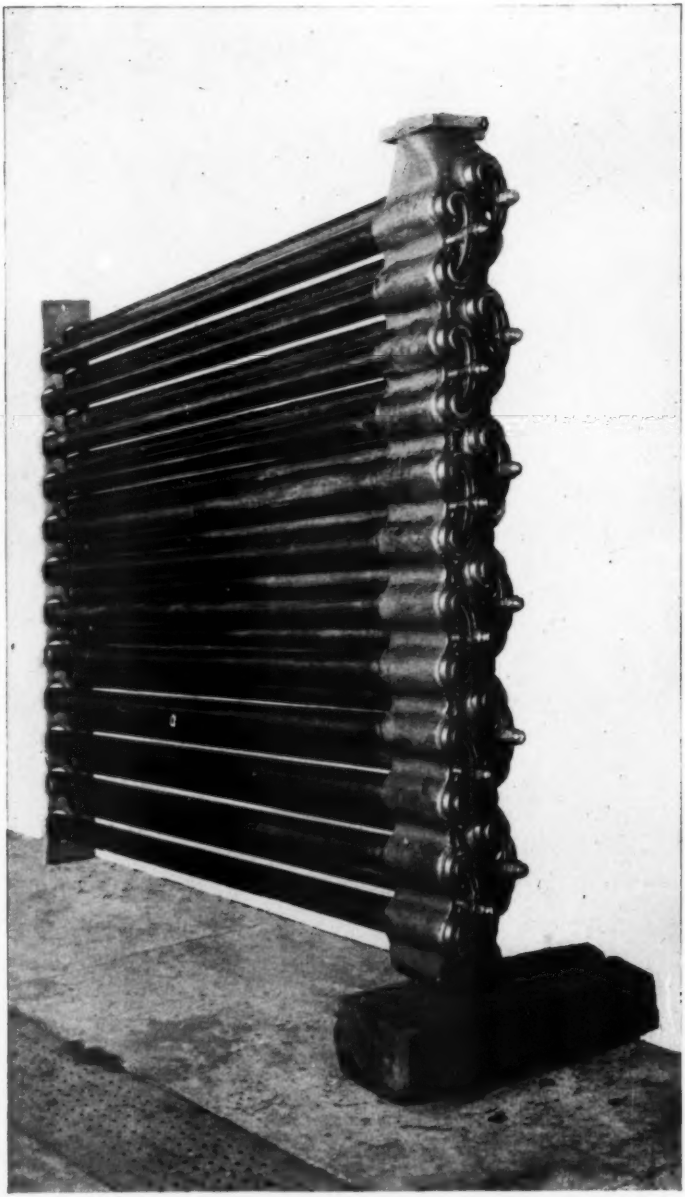
ENGINE DATA.

	H.P.	I.P.	L.P.
Cylinders, number for each engine	1	1	2
diameter, inches	35	57	66
Stroke	48	48	48
Valves, number for each cylinder	1	2	2
diameter, inches	20	22	25
Valve stems, diameter, inches	2	2	2
Piston rods, diameter, inches	8	8	8
axial hole, inches	4	4	5
length	9 ft. 4½ in.		
Cylinder walls, thickness	2	1⅝	1½
Thickness of liners	1½	1⅝	1½
Jacket space	¾	¾	¾
Valve-chest liners	2½	2⅝	2⅝

Crosshead surface, ahead, square feet	690
backing, square feet	510
Connecting rod, between centers, feet	8
Clearance of all pistons, top, inch	1/2
bottom, inch	3/4
Crosshead pin, diameter, inches	9 1/2
length, inches	10 1/2
Crank shaft, number of sections	2
diameter, at eccentric seatings, inches	18 1/2
of axial hole, inches	10 1/2
coupling disc, feet and inches	2- 8
thickness of coupling disc, inches	3 1/2
number of bolts in coupling disc	8
diameter of bolts in coupling disc, inches at center	3 1/2
journals, inches	17 1/2
length of journals, inches	21
forward section, feet and inches	15- 6 1/2
after section, feet and inches	16-10
pin, length, inches	21 1/2
diameter, inches	18
of axial hole, inches	10 1/2
webs, width, inches	20
thickness, inches	11
Thrust shaft, diameter, inches	16 1/2
of axial hole, inches	10 1/2
number of collars on each shaft	12
collars, diameter, inches	27
thickness, inches	2
space between, inches	4
total thrust surface, one engine, square feet	29.335
length, feet and inches	13- 7 1/2
Line shaft, diameter, inches	16 1/2
axial hole, inches	10 1/2
length, feet and inches	14-11
after section, feet and inches	14-11 1/2
diameter of coupling disc, feet and inches	2- 8
coupling bolts in each disc	8
diameter, inches, tapering	4 to 3



SUCCESSIVE STEPS IN FORMING NICLAUSSE HEADERS.



A Niclausse Element.

VALVE DATA, U. S. S. *VIRGINIA*.

Engine, inches 35, 57, 66, 66
Valve gear ..... Stephenson link.
Connecting rod, between centers, feet 8

—	H.P.		I.P.		L.P.	
Eccentricity, inches	5		5		5	
Travel of the valve, inches	10		10		10	
Number and size of piston valves, inches	1, 20		2, 22		2, 25	
Side on which steam is taken	Inside.		Outside.		Outside.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port, inches	3½	3½	3½	3½	3½	3½
Steam lap, inches	2 1/16	2	2 1/16	2	2 1/16	2
Exhaust lap, inches	−⅛	−¼	−⅛	−¼	1/16	⅛
Angular advance, degrees	36	36	36	36	36	36
Steam lead (linear), inch	⅛	1/16	⅛	1/16	⅛	1/16
(angular), degrees and min	8, 27	12, 25	8, 27	12, 25	8, 27	12, 25
Cut-off, inches	37 1/16	33 1/16	37 1/16	33 1/16	37 1/16	33 1/16
decimal of stroke	.775	.707	.775	.707	.775	.707
Mean cut-off	.741	.741	.741	.741	.741	.741
Exhaust release, inches	43 1/16	41 1/16	43½	42	44½	44½
decimal of stroke	.903	.865	.907	.875	.935	.925
Compression, inches	4	3	4 1/16	3½	6½	5½
decimal of stroke	.083	.063	.092	.068	.13	.107
Steam opening, inches	2 1/16	3	2 1/16	3	2 1/16	3
Exhaust opening	Full port	Full port	Full port	Full port.	Full port	Full port.
Velocity of steam (piston speed, 960 feet) feet per minute	7,240	6,160	8,660	7,650	10,250	9,050
Velocity of exhaust, valve ports	5,550	5,280	6,690	6,560	7,860	7,770
cylinder ports	5,600	5,300	6,900	6,770	7,725	7,600

Velocity of exhaust to condenser, 24-inch pipe, feet per minute 7,200

BOILERS.

The *Virginia* has twenty-four Niclausse boilers, arranged in groups of four, in six watertight compartments—three on each side of the middle-line bulkhead and inboard of the coal-bunker bulkheads. In each compartment the boilers are arranged in pairs, each pair consisting of one 16-element boiler inboard and one 15-element boiler outboard, the forward pair facing aft and the after pair facing forward, with an athwartships fireroom space between. Passageways run inboard of the boilers throughout the boiler compartments, and communication is established between opposite compartments and between adjacent compartments on the same side by means of watertight doors. The after boiler compartments communicate with the engine room through air locks.

Each boiler element consists of a vertical forged-steel header, with interior diaphragm, and twenty-four inclined generating tubes, fitted with internal circulating tubes, according to the

Niclausse construction, this arrangement providing 384 generating tubes for each 16-element boiler and 360 for each 15-element boiler, or a total of 8,928. These generating tubes are of steel, seamless drawn, $3\frac{1}{4}$ inches in outside diameter, and No. 6 B.W.G. in thickness, with lanterns integral with the tubes at the header ends, and at the opposite ends, where resting in the rear tube supports, pinched into $2\frac{5}{8}$ inches diameter and capped. The overall length of each tube, inclusive of $9\frac{1}{2}$ inches length of lantern, is 8 feet $3\frac{3}{16}$ inches, and the exposed length 7 feet $5\frac{11}{16}$ inches. The circulating tubes are of No. 21 B.W.G. sheet steel, $1\frac{9}{32}$ inches outside diameter, each formed in two halves, pinched together with folded-over longitudinal joints. The downward inclination to the rear of all tubes is 1 to 10.

An arrangement has been made with the Bureau of Ordnance whereby the compressed air used for the battery is led into the fire rooms and, by means of suitable hose connections, terminating in a pipe with a back-blowing nozzle, may be used to empty the generating tubes of water.

The heating surface of the tubes is 2,291.75 square feet in a 15-element boiler and 2,444.20 square feet in a 16-element boiler. Adding the heating surface of the headers, the total heating surface of a 15-element boiler is 2,320.07 square feet and of a 16-element boiler 2,474.44 square feet. Total heating surface of all boilers 57,534.12 square feet.

The grates in all boilers are 6 feet $8\frac{5}{8}$ inches long, and the width is 8 feet 9 inches for 15-element boilers, and 9 feet $4\frac{1}{4}$ inches for 16-element boilers, making the grate surface for a 15-element boiler 58.78 square feet and for a 16-element boiler 62.86 square feet. The total grate surface is 1,459.68 square feet, and the ratio of heating surface to grate surface 39.41 to 1. The grates are fitted with Navy standard grate bars.

Each boiler has a separate drum, situated directly over the headers and connected to each by means of a double cone nipple, forming Niclausse joints with the bottom of the drum and the top of the header, similar to the joints made by the generating-tube lanterns in the fronts of the headers. These

nipples are each secured by four steel bolts, connecting the bottom of the drum and the top flange of the header. The drums are 10 feet 1½ inches long in a 15-element boiler, and 10 feet 8¾ inches long in a 16-element boiler, and all are 3 feet 6 inches outside diameter and 1½ inches thick. Each drum has the following exterior fittings: Auxiliary feed, main feed, safety valve, water column, try cocks, manhole (12 × 16), surface blow, stop valve, salinometer valve and steam gauge.

The main and auxiliary feed valves are both 1½ inches combination check and stop valves, and on entering the boiler both feed lines unite, discharging into an internal feed pipe of steel 2 inches in diameter and 7 feet 4 inches long.

The safety valves are Ashton twin spring, each 3 inches in diameter, set for 265 pounds. They may be lifted by hand from the fireroom or from the gun deck.

The water column extends from a point 30 degrees from the top to about 50 degrees from the bottom of the drum, and is fitted with the ordinary gauge and a Klinger reflex gauge. The bottom of the gauge glasses is 9½ inches above the bottom of the drum and 4 inches above the lowest try cock, of which there are four of Navy standard pattern, spaced 5 inches apart vertically.

The surface blow is a 1-inch pipe leading down between the boilers to a Y, where it connects with the bottom blow manifold. This manifold has 1-inch pipes leading to the bottom of each header. The surface blow is fitted with the usual scum pan.

The salinometer pipe is 1 inch in diameter and leads from the head of the drum near the bottom.

The stop valve is 4½ inches diameter and takes steam from a tinned-copper dry pipe 3 inches in diameter, extending nearly the length of the drum. This valve may be closed from the opposite fireroom, but not from the deck above.

Three zinc baskets, each holding five plates 12×6×½, are fitted in each drum. A vertical diaphragm with top curved over to the rear is fitted over each header, its base supported by the diaphragm in the header.

Each boiler has one furnace and three balanced vertical swinging doors. The ash-pit doors are vertical, inward swinging, and in case of the bursting of a tube the internal pressure will close them. A fire extinguisher with $\frac{3}{4}$-inch supply pipe is fitted in each furnace, worked from the opposite fireroom. The height of the furnace from the dead plate to the lowest row of tubes is 3 feet. The baffling of the product of combustion is effected by means of two diaphragms in each boiler, the lower situated over the eighth row of tubes, and extending from the headers to within $40\frac{1}{2}$ inches of the rear tube supports, the upper situated over the sixteenth row of tubes, and extending from the rear tube supports to within $30\frac{3}{4}$ inches of the headers. These diaphragms are formed simply by pieces of pipe laid in the spandrels of the generating tubes, and not secured in any manner.

The boiler casings are built up, of inner and outer flat sheets of $\frac{1}{16}$ inch thickness, separated by a corrugated sheet of No. 18 U.S.G. in thickness and $\frac{21}{32}$ inch depth of corrugations. These are lined inside with $2\frac{1}{2}$ inches of magnesia covered by one-quarter inch of millboard. The furnaces are lined with fire brick to the height of the fourth row of tubes at the front and the seventh row at the back.

As originally constructed the lining of magnesia and millboard did not extend below the level of the tubes, but on the first dock trial the side casing became red hot abreast the furnaces, and it was decided to continue the magnesia and millboard down to the bottom.

The tube ends are made accessible by means of eight cleaning doors on the front of each boiler, sliding on a horizontal trolley rail at the top, and bolting at the top and bottom. These doors are removable in turn, when moved to the inboard end of the trolley rail, and the end of any desired tube may be uncovered by removing one door and sliding others along the rail.

There are three smoke pipes, each 9 feet 6 inches inside diameter, and extending 119 feet above the grates.

The boilers were manufactured at the works of the Stirling Company, Barberton, Ohio.

The thrust shaft is fitted with twelve collars, for the horse-shoe thrust block.

The line shafting is in two lengths for each engine, and is supported by two spring bearings.

For each engine there is one section of stern-tube shafting, supported by two stern-tube bearings, and one section of propeller shafting.

SHAFTING.

Each length of shafting is forged solid in one piece, and finished all over, with axial holes drilled from end to end, and also through the crank-pins—these latter used for lubrication.

The crank shafting is in two lengths, each carrying two cranks. A worm wheel for turning the shaft is fitted over the forward coupling.

Total length of shafting, each engine, feet and inches	137- 7½
Propeller shaft, without casing, diameter in strut bearing, inches	17½
length, feet and inches	25- 8½
Stern-tube shaft, after stern-tube bearing, feet and inches	4- 4
forward stern-tube bearing, feet and inches	3- 9
length, stern-tube bearing, feet	36

PROPELLERS.

The *propellers* are three-bladed manganese-bronze, inboard turning, the starboard one being left-handed. They are designed as true screws, and the surfaces of the blades are polished and painted.

Number of blades	3
Diameter, feet and inches	17- 3
of hub, feet and inches	4- 6
Length of hub, feet and inches	3- 6
Helicoidal area, square feet	84.5
Disc area, square feet	233.71
Ratio of pitch to diameter	1.058
Pitch (adjustable one foot either way), feet and inches	18–3

MAIN CONDENSER.

These are two, one in each engine room, shell of steel, water-heads of composition, tubes of Admiralty metal not tinned.

There is a by-pass in the division plate of the waterhead, to allow the circulating pump to discharge directly overboard.

Diameter of shell, feet	7
Thickness of shell, inch	$\frac{8}{8}$
Length over heads, feet and inches	15–10
Tubes, number in each condenser	5,684
diameter, inch	$\frac{5}{8}$
length between tube-sheets, feet and inches	12–10
thickness	No. 18 B.W.G.
space between centers, inch	$1\frac{1}{16}$
Cooling surface, each condenser, square feet	11,969

MAIN AIR PUMPS.

Abaft the condenser in each engine room there is one Blake two-cylinder, vertical, double-acting air pump. The valve discs are composed of three sheets of rolled manganese-bronze, each $\frac{1}{32}$-inch thick, and access is had to these valves by means of a manhole suitably placed.

Diameter of cylinders (two) inches	14
pump (two) inches	32
Stroke, inches	21
Diameter of pump valves in bucket, inches	$5\frac{1}{2}$
Number of delivery valves in each end	15
Area through valve openings, square inches	195

MAIN CIRCULATING PUMPS.

For each condenser there is one centrifugal, double-inlet, circulating pump, driven by a compound, vertical, inverted engine. The capacity of this pump is 12,000 gallons per minute at 266 revolutions, and it may draw from the sea, the main drain or the engine-room bilge. The three suction valves are so interlocked that only one can be open at a time.

Diameter of steam cylinder, inches	11 and 22
Stroke, inches	10
Diameter of pump runner, inches	45
Width of pump runner at periphery, inches	4
at hub, inches	$10\frac{9}{16}$
Radius of inlet nozzle, inches	$14\frac{1}{2}$
outlet nozzle, inches	21
Capacity of pump at 266 revolutions, gallons per minute	12,000

FEED AND FILTER TANKS.

Under each condenser is a combined feed and filter tank of 4,800 gallons capacity. A horizontal partition divides each tank into two compartments, of which the upper is the filter tank, of 1,040 gallons capacity, and the lower the feed tank proper, of 3,760 gallons capacity. Each filter tank is divided into three parts by vertical partitions, and the water flows through all three before reaching the feed tank below. One hundred tons of feed water may be carried in the double bottoms.

FEED PUMPS.

There are two main feed pumps in each engine room. They are Blake, vertical, simplex, center-packed, plunger type, draw water from the hotwell-pump discharge, and discharge into the main feed pipes.

MAIN FEED PUMPS.

Four Blake vertical, simplex, center-packed, plunger type.

Diameter of steam cylinder, inches	14
water cylinder, inches	10
piston rod, inches	2¾
Stroke, inches	12

In each of the six firerooms is one Blake vertical, simplex, center-packed plunger, auxiliary feed pump, bolted to the center-line bulkhead. Each pump may draw water from the feed tanks, from the hotwell-pump discharge, from the main drain, the auxiliary drain, the sea, the floor of its own compartment, and from the bottom blow pipes of all boilers in its own compartment. It may discharge into the auxiliary feed, into the fire main, or overboard in its own compartment.

AUXILIARY FEED PUMPS.

Six Blake vertical, simplex, center-packed, plunger.

Diameter of steam cylinder, inches	11
water cylinder, inches	8
piston rod, inches	2¼
Stroke, inches	12

FEED-WATER HEATER.

In each engine room is one feed-water heater, containing 950 square feet of heating surface in 852 tubes. Auxiliary exhaust steam is the heating agent. The hotwell pump delivers water through the heater to the feed-pump suction.

HOTWELL PUMP.

In each engine room is a hotwell pump which may draw water from either air-pump suction pipe, from hose connections on the side of the ship above the water line, and from the main and the reserve feed tanks. It may discharge into the main feed suction line, either directly or through the feed-water heater, or into the main or reserve feed tanks, or auxiliary feed line.

FIRE AND BILGE PUMPS.

In each engine room is one Blake, vertical, simplex, fire and bilge pump 12×10×12. These pumps draw from the sea, the main drain, the auxiliary drain, the drainage manifolds, the engine-room bilge and the crank pits. They discharge to the fire main or overboard.

Diameter of steam cylinder, inches	12
water cylinder, inches	10
piston rod, inches	2¼
Stroke, inches	12

There is placed in the port engine room a distiller circulating pump, Blake, of 350 gallons capacity. It draws water from the sea, and discharges into the flushing system and the distiller circulating pipe, and through the latter into the flushing system or overboard.

AUXILIARY CONDENSERS.

In each engine room there is one auxiliary condenser having 601.4 square feet of cooling surface measured on the outside of the tubes, which are 552 in number, of Admiralty metal, not tinned, 6 feet 10⅛ inches long, ⅝-inch outside diameter, and No. 18 B.W.G. in thickness.

In the dynamo room is one auxiliary condenser having 1,200 square feet of cooling surface. It has 1,001 tubes similar to the above, but 7 feet $6\frac{1}{8}$ inches long.

Each condenser is fitted with a combined air and circulating pump, that for the dynamo condenser taking steam from the bottoms of the dynamo separators. The dimensions of these pumps are, for the engine-room condensers, $18\times10\times10\times12$, and for the dynamo-room condenser $12\times14\times14\times12$.

FORCED DRAFT.

The closed fireroom system of forced draft is used, there being two Sturtevant blowers in each fireroom.

Number of blowers	12
Type of engine	Two-cylinder, vertical.
Diameter of steam cylinders, inches	6
piston rods, inches	$1\frac{1}{16}$
fan, inches	66
Stroke, inches	5

EVAPORATING PLANT.

There are four evaporators, of Bureau type, each having a capacity of 4,750 gallons per day, or a combined capacity of 19,000 gallons per day. Each evaporator has ninety-six 2-inch brass tubes with a tube-heating surface of 240 square feet. There are three distillers, of Bureau type, each having a capacity of 6,333 gallons per day, or a combined capacity equal to that of the evaporators. Each distiller has 187 tinned-brass tubes with a cooling surface of 127.45 square feet. Circulating water for the distillers is supplied by the distiller circulating pump through the flushing system, or may be drawn from the fire main. The evaporators are situated on the berth deck at the after end of the boiler-room hatch, and the distillers are in a trunk immediately above them.

ELECTRICAL PLANT.

There are two generators of 100 kilowatt and six of 50 kilowatt, all at 125 volts. These are situated in four dynamo rooms, two on each side of the fore-and-aft bulkhead, two at the level of the hold and two above these.

The 50-kilowatt machines are 6-pole, 400-ampère, designed to run at 400 revolutions per minute while the 100-kilowatt generators are 8-pole, 800-ampère, and run at 350 revolutions per minute.

The generators are of Thresher manufacture, while the engines are built by the Forbes Co.

THE HEATING SYSTEM.

Radiators and heaters throughout the ship are arranged in circuits, each circuit being so connected that it can be operated independently of the others. The circuits are, in turn arranged in groups, each group, having a receiving manifold and a drainage manifold. Steam is taken from the auxiliary steam pipe at the receiving manifold. Here there is a stop valve for each circuit, and a stop valve, an adjustable reducing valve, and a steam gauge at the connection with the auxiliary steam pipe. The drainage manifolds have a stop valve and a check valve for each circuit. Each manifold discharges through a trap, with a by-pass to the feed tanks, the auxiliary condenser or the bilge. By this arrangement each circuit can be entirely isolated, and water from other circuits prevented from backing up into it.

REFRIGERATING PLANT.

An Allen dense-air ice machine of three tons capacity is installed forward on the port side of the berth deck. The refrigerating rooms, four in number, and the ice-making boxes, are adjacent to it. Cooling pipes also lead to the scuttlebutt.

MACHINE SHOP.

This is located on the upper platform between the after fireroom and the engine room, and is divided into two parts by the fore-and-aft watertight bulkhead. Communication between the two is established by a watertight door. The following tools are installed: One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers; one 14-inch screw-cutting back-geared lathe, to take 4 feet between centers; one column-shaping machine of

16-inch stroke and 24-inch traverse; one back-geared drilling machine, to drill up to 1½ inches, 14 inches from edge of work; one 16-inch sensitive drill; one No. 1 universal milling machine; one combined hand punch and shears, with 6-inch shear blades, capable of cutting ¾-inch round iron, shearing ⅝-inch steel plate, and punching ⅝-inch holes in ⅝-inch mild-steel plates; one emery grinder with two wheels 12-inch in diameter and 2-inch in face; one 30-inch grindstone; six bench vises. The machine tools are driven by one electric motor in the machine shop.

STEAM PIPING.

The main steam pipes, of seamless-drawn steel, are arranged symmetrically in two systems, one for each side of the ship, each system being independent of the other except for a 9-inch cross-connecting pipe uniting the systems near the main engines. The system for each side of the ship is as follows:

An 8-inch branch pipe leads from each compartment or nest of four boilers. The branches from the forward two compartments unite in a 10½-inch pipe, and this, in turn, unites with the pipe from the after compartment into a 12-inch pipe to the engine room. Stop valves are placed on each branch, and on the main pipe just forward of each junction. The 8-inch branch pipes take steam directly from the auxiliary steam pipe, and so, indirectly, from the boilers. Abaft the engine-room bulkhead the 12-inch pipe has, in the order named, a bleeder connection to the condenser, a 9-inch cross connection to the other side of the ship, a main stop valve, a connection to the auxiliary steam pipe, a separator, an expansion joint, and the main-engine throttle valve.

AUXILIARY STEAM PIPING.

This system consists of two main arteries 8 inches in diameter, extending fore-and-aft through the firerooms, one on each side of the ship, having a pipe 6 inches in diameter, extending from the after firerooms to the engine rooms with branches to the auxiliary machinery, and a 6-inch extension from each

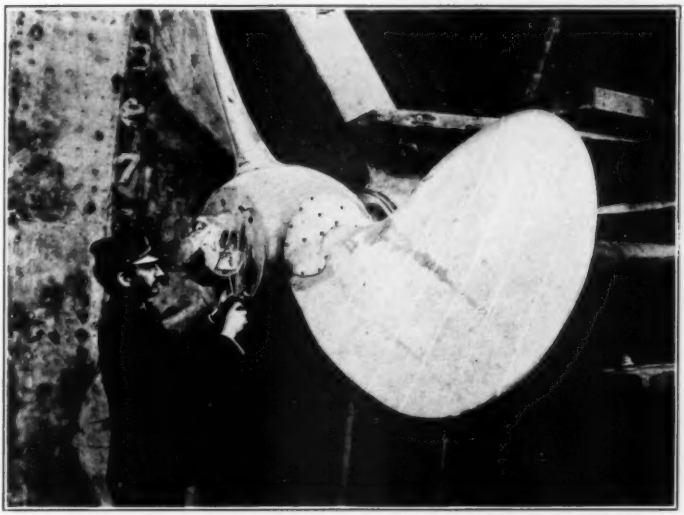
main artery to the dynamo rooms and the forward-deck machinery, these extensions cross connected in the forward fire-rooms. Each main artery has a 6-inch branch with stop valve for each athwartships pair of boilers. The 6-inch branches subdivide, each into two 4½-inch branches, and each 4½-inch branch connects with a boiler stop valve. The main and auxiliary steam pipes are connected in each engine room.

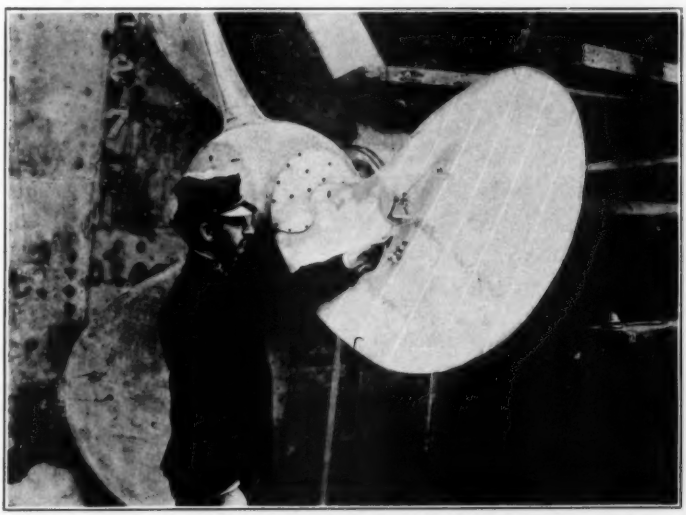
AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe of copper is connected to all of the auxiliary machinery. It has valves to direct the exhaust steam into either auxiliary condenser, either main condenser, either low-pressure receiver, either feed-water heater or into the after escape pipe. The dynamo engines exhaust either into the dynamo condenser or into the auxiliary exhaust pipe.

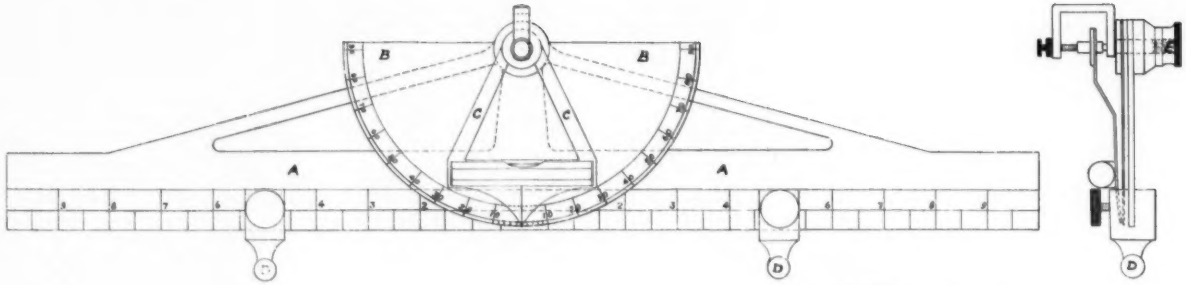
OFFICIAL TRIALS.

The official preliminary trials of this vessel were held November 21–23, 1905, a full account of which will appear in the next issue of the JOURNAL.





THE WINGATE PITCHOMETER.



PITCHOMETER

DESIGNED BY C. M. WINGATE,

WARRANT MACHINIST U.S.N

NOTES.

THE WINGATE PITCHOMETER.

The following notes were furnished by Lieutenant Commander T. W. Kinkaid, U. S. N.

The pitch-measuring device herewith illustrated is the invention of Warrant Machinist Clarence M. Wingate, U. S. N. It can be conveniently used when a vessel is in drydock, but is also available when the propeller to be measured lies on the shop floor.

The pitchometer is based on the formula: Pitch = circumference × tangent of the angle made by the helix with the circumference. As commonly used, the ship being in drydock, with the blade horizontal, the reading taken is the complement of the angle required by the formula. On the back of the blade of the instrument are set forth a column of angular readings and the corresponding pitches for a diameter of one foot. For measurements taken at different diameters it is only necessary to multiply the tabulated pitch by the diameter employed, taken in feet.

When taking readings it is convenient to have parts of helices drawn on the blade with chalk at various radii. The helices are drawn with the aid of a flexible batten through two or three points located on the blade by means of straight edge and square. The instrument is capable of indicating whatever variation of pitch may exist at different points of the helix.

Figure 1 illustrates the construction of the pitchometer. The pendulum index should be plumb when the reading is taken; and to give information as to this fact a spirit level is affixed to the pendulum. The graduated quadrant is pivoted, and may be clamped in any angular position desired. As, in docking, the propeller shaft is seldom level, it is necessary as

a preliminary step to remove the protecting cap from the propeller nut and apply the two feet of the instrument to the plane surface of the boss, at the same time bringing the quadrant to a zero reading, in which position it is clamped.

THE DÜRR WATER-TUBE BOILER IN H.M.S. *ROXBURGH*.

When the Admiralty decided, on the recommendation of the Water-Tube Boiler Committee, to fit the cruisers of the *Devonshire* class with four different boiler systems, the Dürr boiler was included, partly because of its success in the German Navy and partly as a result of the tests made by the Committee with an experimental installation in the small cruiser *Medusa*. It was therefore fitted in H.M.S. *Roxburgh*. The steam trials of this vessel, completed in May last, gave satisfactory results, and since then the vessel has been put in commission, and considerable interest will be taken in the performance of the steam generators in service.

All the ships of this class have approximately one-fifth cylindrical and four-fifths water-tube boilers; and in the case of the *Roxburgh* the ship, engines and cylindrical boilers were constructed by the London and Glasgow Engineering and Iron Shipbuilding Company, Limited, and the Dürr water-tube boilers were built and fitted on board by the Fairfield Shipbuilding and Engineering Company, Limited, who are the sole licensees for this type of marine boiler in Britain and the Colonies. There are seventeen water-tube boilers of this type fitted in the *Roxburgh*, supplying steam for 16,000 indicated horsepower, the remaining power, 5,000 indicated horsepower, being developed from steam generated in six cylindrical boilers working under Howden's forced-draft system.

The Dürr boilers, to which we more particularly wish to refer, are arranged in three separate compartments, and have a collective heating surface of 41,600 square feet, a superheating surface of 2,245 square feet, and a grate area of 1,085 square feet, with a working pressure of 220 pounds per square inch.

The construction of the boiler is very similar to the most recent generators of this type fitted in the German Navy. Several modifications, however, have been introduced in the design, resulting from experience gained in the trials of the *Medusa*. The boiler consists essentially of a large steam collector, to which is riveted a vertical water chamber extending across the whole front of the boiler. The tubes, which are of the Field type, with internal circulating tubes, are connected to this water chamber at the front of the boiler, while at the back they simply rest in a lattice-like wall built up of steel flat bars, so that each tube is free to expand independently. Superheating tubes, with similar internal tubes, are also fitted to the upper part of the steam collector, to ensure that absolutely dry steam passes the boiler stop valve. Tube-hole doors are arranged opposite each tube in the water chamber, and also in the steam collector. The water chamber is built of mild-steel plates, welded throughout, and supported by means of screwed stays between the tube holes. It is divided internally by a vertical division plate, through which the stays are also screwed, and to which the inner circulating tubes are connected. The feed water from the steam collector passes through the front part of the water chamber into the inner circulating tubes, leaves the same at the back end, and returns through the boiler tubes to the back part of the water chamber, and from thence it rises to the steam collector.

The two lowest horizontal rows of boiler tubes are $3\frac{1}{2}$ inches in external diameter, and the remaining sixteen rows are $3\frac{1}{4}$ inches in external diameter. The front end of each is staved up in an hydraulic press, and turned in a lathe slightly conical to make the joint with the steam collector, the axis of the cone being set at a slight angle to the axis of the tube in order to get the required inclination of the tubes to the water chamber. The back ends of the tubes are reduced in diameter, screwed, and closed up by means of a removable forged-bronze cap nut, jointed against a steel collar permanently fixed to the screwed end of the tube. The tube joints in the water chamber are made by forcing the tubes in their coned seat with a small

hydraulic ram connected to a hand pump, the pressure required being about 2,500 pounds per square inch. With this pressure the tubes are pressed into the tube holes about $\frac{5}{16}$ of an inch, after having been previously bedded, by light blows on a wooden plug, in the cone. The joint surfaces are thinly coated with a graphite composition, which lessens the friction when pressing in the tube, and facilitates removal again later when required. The tubes are readily withdrawn when necessary by means of a screwed rod passed through the tube; and if, by repeated forcing into their seat, the safety collar which is provided at the end of each tube comes so close to the tube plate as to prevent further entry, the tube cone joint may be readily expanded with an ordinary expander, a heavy steel gauge ring, bored to the size of the standard cone, being first placed over the joint surface of the tube.

The inner circulating tubes are $1\frac{3}{4}$ inches in diameter; they are made from thin, folded steel sheets, and held in position at the water-chamber division plate by a bell mouth loosely riveted to the end of the tube to form a hinge joint, which rests on two small clips. The back end of the tube is held in position central with the tube by three springs riveted to the tube.

The two vertical rows of boiler tubes on each side of the boiler are bent right and left in such a way that they form a complete water wall, giving the side casings an effectual protection from the furnace gases.

The superheater tubes connected to the steam collector are $2\frac{3}{4}$ inches in external diameter, with inner tubes $1\frac{1}{2}$ inches in diameter, and are jointed and held in position in a similar manner to that of the water tubes already described. The steam enters the inner tube through slots at the top of the outer division casing, and returns by the outer tube to the inner division casing, the stop-valve orifice being connected thereto.

The gases are baffled on their passage between the tubes by plate baffles resting on the tubes and inserted through the lattice wall at the back of the boiler; the lower row of front

baffles are held in position by a thin wire rope passed through a row of stays bored hollow for this purpose, and having the end hooked over the nuts of the tube-hole doors below.

The tube-hole doors are of forged steel, circular in form, with joint in the front plate of the water chamber coned in a similar manner to that of the tubes, and, the doors being entered from the inside of the water chamber, the steam pressure tends to keep the cone tight in its seat. The diameter of the door openings is just sufficient to pass the tubes through; and, when placing the tube-hole doors in position, the doors for tubes $3\frac{1}{4}$ inches in diameter are passed through the larger openings opposite the $3\frac{1}{2}$-inch diameter tubes, and the doors for the latter diameter tubes are entered through a few oval openings. Each door has a screwed end forged solid with it, and a nut, bearing against a light steel cap which covers the outside of the opening, enables the door to be tightened up on the cone joint.

The casings are substantial, yet light in construction, and the large openings through the diamond-shaped lattice work at the back of the boiler, on which the tube ends rest, admits of brushes, or lances for compressed air, being inserted for sweeping the tubes. In front of this lattice work, cover plates, easily portable, and consisting of two thin steel plates with asbestos between, are fitted; these prevent the hot gases passing through the lattice work and ascending in the space between the lattice work and external back end doors of casing.

Air tubes with portable caps, and air doors with pull rods led to the front of the boiler, are provided for admitting a supply of air above and below the fires, respectively, at the back of the furnace.

The water chambers and steam collectors are supported on columns of channel section forming part of the lower front casing, and are also tied to the framework forming the back support for tubes.

Two of the boilers were tested under steam on shore before being fitted on board, at rates of combustion varying from

18 pounds to 30 pounds of coal per square foot of grate per hour. These trials were very satisfactory, the steam produced being extremely dry.

During the steam trials* of the vessel the boilers worked well; the stoking was carefully and systematically carried out at regular time intervals, clocks in each stokehold being arranged with special dials for this purpose, and from beginning to end of the trials no trouble was experienced with the Dürr boilers.

On the trial at one-fifth power the Dürr boilers evaporated, from and at 212 degrees Fahrenheit, 8.85 pounds of water per pound of coal, the fuel consumption for all purposes being 2.09 pounds per indicated horsepower per hour. On the 30-hours trial at 15,000 indicated horsepower the evaporation was 8.4 pounds per pound of coal, and the mean fuel consumption for all purposes 1.99 pounds per indicated horsepower per hour. On the full-power trial, when the main engines indicated 21,857 indicated horsepower, the Dürr boilers worked with an air pressure of 1 inch, and the cylindrical boilers with 1⅛ inches, the mean evaporation being 7.87 pounds per pound of coal, and the coal consumption 2.3 pounds. The Dürr boilers burned 39.5 pounds of coal per square foot of grate, while the cylindrical boilers consumed 27 pounds per square foot of grate.

—"Engineering."

TRIALS OF 500-B.H.P. DIESEL OIL ENGINE.

Abstract from report of Chief Engineer of British Engine, Boiler and Electric Insurance Co.

The engine was a three-crank inverted vertical, with three single-acting cylinders, numbered 54, 55 and 56, No. 54 being above the idle end of the crank shaft. Each cylinder was 22.05 inches (560 millimeters) diameter, with a piston stroke of 29.53 inches (750 millimeters). The normal speed was 150 revolutions per minute.

The valves were placed in the cylinder covers as usual, and were actuated by levers driven by cams on a horizontal shaft,

* See "Engineering," vol. lxxix, page 681.

which in turn was driven by a vertical shaft and bevel gear from the idle end of the crank shaft. The cylinders, cylinder covers and exhaust valves were water-jacketed, but the pistons were not. The engine drove a dynamo carried upon a prolongation of the crank shaft.

The air for pulverizing the oil and spraying it into the cylinders was compressed in an independent pair of three-stage vertical air compressors worked by a two-throw crank shaft, belt driven by a motor receiving current from the dynamo upon the engine crank shaft. The air compressors, therefore, though essential to the working of the engine, were not in this case parts of the engine, and in calculating the mechanical efficiency of the engine from the dynamo output and the indicator diagrams this fact should not be lost sight of. Had the compressors been driven directly by the engine, the difference between the work put into the dynamo, which is the brake horsepower, and the indicated horsepower, would have been increased by the power required to compress the air.

The areas of the compressor pistons were:

102.40 sq. in. (660.5 sq. cm.),
32.30 sq. in. (208.1 sq. cm.),
8.79 sq. in. (56.75 sq. cm.),

and the stroke 7.087 in. (180 millimeters), the speed, when compressing to about 64 atmospheres, being about 160 revolutions per minute.

The arrangement, which, of course, is somewhat wasteful, was adopted to meet special conditions. In ordinary cases it is proposed for engines of this size to use Riedler compressors driven from the engine crank shaft by connecting rods or belts.

The dynamo was 12-pole, continuous current, shunt wound, by Lahmeyer & Co., rated to give 450 kilowatts at 550 volts when running at 150 revolutions per minute. The efficiencies given by the makers are—

	kw.	kw.	kw.	kw.	kw.
At	112	225	337	450	562
Efficiency about . .	.88	.925	.935	.94	.935

and these figures have been adopted in calculating the brake horsepower of the engine corresponding to the measured output of the dynamo.

The power was absorbed by iron wire resistance coils, and the load regulated by appropriate switches. The motor for driving the air compressors was 6-pole, shunt wound, continuous current, by the same makers, rated to give 75 B.H.P. at 630 revolutions per minute.

The calculated efficiencies given by the makers are:

At	Full load.	Three-quarter load.	Half load.	Quarter load.
Efficiency . .	90.5	89	86	76.5m.

DETAILS OF THE FOUR TRIALS MADE.

Four trials were made, the results of which are shown on the accompanying tables.

The first, a preliminary trial, intended to be at full load, but actually a little below; the second at full load, the third at half load, and the fourth with no external load, the engine driving the air compressors only, and, of course, the dynamo and motor which transmitted the power to them.

With respect to the figures in the table, the following explanations should be read:

Line 4.—The diameter of the cylinder of No. 56 engine was gauged. The diameters of the other two were taken from the drawing.

Line 6.—The revolutions were recorded by an engine counter, and the speed indicated by a tachometer.

Line 7.—The water for the jackets was supplied from the town's main, and measured through a water meter which was said to have been recently calibrated.

Line 9.—The discharge pipes from the jackets were conducted to a common pipe discharging into a drain. The same thermometer was used for measuring the temperature of inlet and discharge.

Line 11.—The temperature of the exhaust was measured close to the engine by a mercury thermometer passing through

a gland in the exhaust pipe, with compressed nitrogen above the mercury to prevent the latter boiling.

	—	Trial I.	Trial II.	Trial III.	Trial IV.
1	Date	13th Feb.	14th Feb.	14th Feb.	14th Feb.
2	Time	3'55 p.m. to 5'55 p.m.	9'17 a.m. to 11'12'15	11'21'15 to 1'30'15	2'1'30 to 2'59'30
3	Duration, minutes	120	115.25	129	58
4	Diameters of cylinders, inches	22.05	22.05	22.05	22.05
5	Stroke of pistons, feet	2.4605	2.4605	2.4605	2.4605
6	Revolutions per minute	150.16	152.8	150.3	150.2
7	Jacket water per minute	166.8	169.85	157.85	140
8	Initial temp. jacket water, °F.	46.3	46.3	46.3	46.3
9	Final temp. jacket water, °F.	125	127.4	104.6	82.1
10	Temperature of outside air, °F.	48	48	48	48
11	Temp. of exhaust gases, °F.	783	806	496	275
12	Analysis of exhaust gases, CO_2	5.6	6.8	3.2	...
	N	42.9	51.3	21.5	...
	Air	51.5	41.9	75.3	...
13	Oil used, pounds	390.3	398	221.1	44.23
14	Oil used per hour, pounds	195.1	207.2	102.8	45.76
15	Blast pressure, atm.	61.5	66.3	50.9	35
16	Maximum pressure shown by indicated diagrams, pounds per square inch	510 515 525	515 525 500	490 480 505	485 480 500
17	M.E.P., pounds per sq. in.:				
	On first piston, No. 54	82.9	80.7	51.6	23.2
	On second piston, No. 55	92.3	93.9	52.6	20.1
	On third piston, No. 56	110	115.6	64.8	33.9
18	Average in the three cyls.	95.05	96.7	56.33	25.4
19	Indicated horsepower	609.3	634.8	363.6	163.3
20	Oil per I.H.P. per hour, pounds.	0.3202	0.3264	0.2828	0.28
21	Output of dynamo, kw.	333	352	168.2	22.24
22	Brake H.P. of engine	475.5	502.5	245	546
23	H.P. absorbed in friction	133.8	132.3	118.6	108.7
24	B.H.P. / I.H.P.	0.78	0.805	0.675	0.334
25	Oil per kilowatt hour, pounds.	0.586	0.5886	0.6113	2.057
26	Oil per brake H.P. per hr., lbs.	0.4103	0.4123	0.4196	0.838
27	Power absorbed by motor, kw.	38	41	31.6	23.6
28	H.P. given out by motor	44.8	48.3	36.8	26.2
29	I.H.P. in compressor cylinders.	36	40	28.8	18.2
30	Power absorbed in belt and compressors, H.P.	8	8	8	8
31	Estimated brake H.P. of engine deducting mean of lines 26 and 27 from line 21, B.H.P.	435.1	458.7	213.8	324
32	Estimate of mechanical efficiency of engine if pump had been driven by it	0.715	0.723	0.588	0.198
33	Oil per brake H.P., pounds	0.444	0.451	0.481	1.415

The above table gives details of the four trials made.

All these observations ware taken at intervals of ten minutes.

Line 13.—The oil used was from Galicia, costing 100 francs per 1,000 kilos. (£4 per ton), delivered at the works. There is considerable doubt about the calorific value of the oil.

The oil used was very carefully weighed in order to remove any doubts which might be raised by the figures in the heat account of trial III.

Lines 17–19.—The mean effective pressures were calculated from indicator diagrams taken at intervals of fifteen minutes.

It will be seen that the mean pressures in the different cylinders differed considerably.

To afford some check upon the dynamo efficiencies given by the makers, the C^2R losses in the armature and magnet coils and in the shunt regulator were measured. With the full load of 350 kilowatts these were as follows:

	kw.
C^2R loss in armature brushes, etc., ...	8.9
shunt coil, ...	3
shunt regulator resistance, . . .	2.5
Total, ...	14.4

Assuming the iron losses and friction (which could not be measured) to be equal to the above, the total losses would amount to about 28.8 kilowatts, giving an efficiency of 92.4 as against 93.5 claimed by the makers.

Line 22.—The brake horsépower given in this line is at each load the measured output of the dynamo in horsepower divided by the coefficient of efficiency given by the makers for that load. It includes the power absorbed by the motor and air compressor.

Lines 27 and 28 give the kilowatts supplied to the motor which drove the air compressors and the horsepower given out by it.

The difference of 8.8 horsepower between this figure and the figure 44.8 in the first column of line 25 represents the power absorbed by the driving belt and the mechanism of the compressors.

HEAT ACCOUNTS.

The high percentage of heat unaccounted for on the second trial may be due to imperfect combustion, especially during the earlier part of the trial, when for a short time there was a little smoke from No. 56 cylinder, and the excess of heat accounted for on the third trial is most likely due to overestimation of the weight of the exhaust gases. This weight varies inversely as the percentage of CO_2 in the gases, and any leakage of air into the gas sample reduces this percentage and consequently unduly increases the calculated weight of the gases.

HEAT OIL ACCOUNTS FOR WHOLE TIME PER ONE POUND OF OIL AND PER I.H.P. PER MINUTE.

—	Trial I.		Trial II.		Trial III.		Trial IV.	
	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.
To calorific value of one pound of oil..	20,050	100	20,050	100	20,050	100	20,050	100
By heat equivalent of work done.....	7,944	39.6	7,794	38.9	8,998	44.9	9,078	45.3
By heat carried off in jacket water...	4,070	20.3	4,110	20.5	5,300	26.4	6,570	32.8
By heat carried off in exhaust gases..	7,030	35.1	6,056	30.2	7,200	35.9	4,402	21.9
By radiation and error	1,006	5	2,090	10.4	1,438	—7.2		
	20,050	100	20,050	100	20,050	100	20,050	100
To calorific value of oil used per I.H.P. per minute	107	100	109	100	94.5	100	93.6	100
By heat equivalent of work done	42.4	39.6	42.4	38.9	42.4	44.9	42.4	45.3
By heat carried off in jacket water...	21.7	20.3	22.3	20.5	25	26.4	30.6	32.8
By heat carried off in exhaust gases..	37.5	35.1	32.9	30.2	33.9	35.9	20.6	21.9
By radiation and error.....	5.4	5	11.4	10.4	—6.8	—7.2		
	107	100	109	100	94.5	100	93.6	100

HEAT ACCOUNTS FOR TRIALS I, II AND III, PER ONE POUND OF OIL, AFTER TEMPERATURE CONDITIONS HAD BECOME NEARLY CONSTANT.

—	Trial I.		Trial II.		Trial III.	
	B.T.U.	Per cent.	B.T.U.	Per cent	B.T.U.	Per cent.
To calorific value of one pound of oil.....	20,050	100	20,050	100	20,050	100
By heat equivalent of work done.....	7,886	39.3	7,893	39.4	8,949	44.7
By heat carried off in jacket water.....	4,440	22.1	4,353	21.8	4,790	23.9
By heat carried off in exhaust gases.	7,090	35.4	6,100	30.4	6,950	34.6
By radiation and error.....	634	3.2	1,704	8.4	—639	—3.2
	20,050	100	20,050	100	20,050	100

During the two full-load trials the temperature of the exhaust and of the discharge from the jackets continued to rise

for some time after commencement of each trial, while during the half-load trial both temperatures fell. The inference is that during the earlier parts of the two first trials the temperature of the cylinder walls and pistons was increasing by absorption of heat from the gases and during the third was falling by imparting heat to the gases. To show the effect of these exchanges of heat, heat accounts have been calculated for the periods during which the temperature conditions were fairly constant. The effect upon the oil consumptions and thermal efficiencies was practically negligible.

The distribution of the oil in equal quantities to the cylinders of a multiple-cylinder engine presents difficulties which have not yet been overcome.—"Page's Weekly."

GOLDSCHMIDT'S SYSTEM OF ALUMINO-THERMICS.

(Extract from the Report of the Committee on Science and the Arts of the Franklin Institute.)

The subject of this report has been repeatedly brought to the attention of our members both at stated meetings of the Institute and of its Chemical Section. Scientific and popular accounts, profusely illustrated with experiments, of Dr. Hans Goldschmidt's inventions have been given by members of the Institute as well as by representatives of the inventor, so that it would seem quite superfluous here to enter into a detailed description of the process or to enumerate all the various applications that have been devised or proposed by the inventor.

The sub-committee further deems it advisable to confine its report to "Alumino-Thermics" proper; that is to say, to that part of Dr. Goldschmidt's work which concerns the production and utilization of very high temperatures by means of mixtures of aluminum and with a metallic oxide, such as oxide of iron. Not only have his efforts along these lines led to the most important practical results, but they embrace all that (in the sub-committee's opinion) is novel in the process.

Of the several patents granted to the inventor by the United States Patent Office between March 16, 1897, and August 4,

1903, three relate to the joining of metal pieces or welding and one to a process of manufacturing homogeneous metal castings.

The heat-producing material known as Thermit, is now manufactured by Dr. Goldschmidt at his Essen works on a considerable scale. It consits of an intimate mixture of finely divided aluminum and oxide of iron (or other metallic oxide), the ingredients being present in chemical proportions—

$$2Al + Fe_2O_3 = 2Fe + Al_2O_3.$$

When this thermit is ignited in one spot, the reaction proceeds with marvellous rapidity through the entire mass without any further supply of external heat. Owing to the high temperature of ignition of thermit, the reaction is started either with a bit of burning magnesium ribbon, or, better, by the aid of a special ignition powder, consisting of peroxide of barium and aluminum powder.

The extraordinary thermal effects of the reacting mass (its temperature is said to approach 5,400 degrees Fahrenheit,) are due to the sudden disengagement of a vast quantity of heat in a relatively small volume of material. The only products are metallic iron and oxide of aluminum, which retain practically the entire heat of the reaction, and, being liquid, separate into two layers—the metal below and the alumina-slag on top.

As already stated, the principle use of thermit is in joining metal pieces, especially of iron and steel.

Thermit welding has been successfully applied during the past four years in a great number of cases, for some purposes on an extensive scale. Conspicuous among these uses of thermit is the welding of street-railway rails at the joints for the construction of continuous track. Many thousands of such welds have been made in England and Germany, and, while the method has not so far found introduction on a large scale in this country, it has been adopted and employed for repairs of the more costly and complex parts of tracks, such as crossings, switches, etc., with excellent results, as one of the members of the sub-committee can attest from personal experience.

Special mention should be made also of an advantage in thermit welding, in that it dispenses with all bulky equipment, for it requires only a few simple appliances in addition to the thermit itself. Of the latter, the Goldschmidt works now supply "welding portions," each containing the quantity required for one weld. In the case of rail ends, the weld is made by bringing the molten iron resulting from the reduction of the thermit in direct contact with the rails, so that it will adhere to the parts to be joined, and thus unite them.

Another method of thermit welding, applicable to wrought-iron pipes and angle iron, depends entirely upon the heat obtainable from thermit, the reduced iron not being allowed to adhere to the welded parts. To this end it is only necessary to apply the molten thermit mass in such a way that the slag (alumina) first comes in contact with the cool surface of the articles so as to form a protecting layer, and thus prevent the metal which follows from adhering. While the ends of the pieces are brought to the welding temperature, the weld is produced by application of pressure, but without disturbing or changing the section of the welded parts.

Thermit welding has also proved most valuable as a means of repair in break-downs of machinery. Many cases have been reported where otherwise exceedingly difficult, if not impossible repairs were made by its means, and so far as the sub-committee is able to ascertain, with remarkably uniform success. It is to be noted in this connection that the repairs which have been made by the aid of thermit include smaller operations, such as replacing a tooth in a steel wheel, as well as the welding of very large pieces, like the stern posts and crank shafts of trans-Atlantic steamships.

Still another ingenious application of the thermit reaction is in preventing the formation of cavities in large steel castings. A box of specially-prepared, so-called anti-piping thermit, attached to an iron rod, is pushed into the ingot, just as a crust is beginning to form. The heat produced is sufficient to keep the metal liquid where the piping occurs and allows more liquid metal to be poured in to fill the cavity.

It is the opinion of the sub-committee that the invention of thermit and its application in welding and casting metals constitutes a great step in advance in the production and utilization of high temperatures. While it is recognized that Dr. Goldschmidt was not the first one either to observe or to apply the heat liberated in this reduction of metallic oxides with aluminum, he must be credited with having developed the manufacture of thermit, with the practical application of the enormous heat energy stored in this product, and with extraordinary perseverance and resourcefulness in bringing these inventions to commercial success and thereby creating a new industry.

The chief obstacle to the more extended use of thermit in this country has been the rather high price of the article with the added import duty. It is to be expected, however, that this barrier will be removed as soon as thermit can be supplied by an American factory.

The sub-committee unanimously recommends the award of the Elliott Cresson Medal to Dr. Hans Goldschmidt for his inventions in "Alumino-Thermics," by the Franklin Institute.

THICK FIRES.

It is the prevailing opinion of some owners of power plants that it is necessary, when a boiler is worked at a high rate, to maintain correspondingly heavy fires. It is argued that thin fires are well enough for slow rates of combustion, but as the call for steam increases it must be met by an increased thickness of the bed of coal on the grate. Where heavy fires are carried it is a common thing for the fireman to shovel in all the coal he can conveniently supply, going so far as nearly to fill the opening at the firedoor, leaving little if any room for a future supply until that already in has been pushed back to make room for more. The ordinary fireman is likely to favor this method, for the reason that he can introduce large quantities at a firing, and afterwards he is not obliged to give the fires much attention for perhaps an hour's time, when he will again fill the furnace full in the same manner as before.

This method of firing with most of the high-class bituminous coals in use in the Eastern States requires from time to time the use of the slice bar for breaking up the bed of coal. Whatever necessity there may be, according to the popular idea, for carrying heavy fires, so far as the amount of work involved is concerned, it is really more laborious for the fireman than it would be to keep the fires comparatively thin and supply small quantities of coal at each firing. An explanation of the favor which this method receives may be that the class of labor generally employed for firing considers the muscular effort required much less of a task than the more frequent and careful attention which is required when the fires are kept at medium thickness.

So far as the engineering merits of thick and thin fires are concerned, the fact is more capacity can be obtained from a boiler when a fire of medium thickness is carried and proper attention is given to its condition than can be realized by any system of management when the fires are exceedingly heavy. Advocates of thick fires, who take the ground that they are a necessity when boilers are forced, are entirely mistaken. As to the economy of the two, some persons maintain that heavy fires give the more economical results, but this is questionable. Valuable information on this point is afforded by two evaporative tests made by Mr. George H. Barrus. They were made on a 72-inch return-tubular boiler having 100 3½-inch tubes 17 feet long. The heating surface amounted to 1,642 square feet and the grate surface to 36 square feet, the ratio of the two being 45.6 to 1. On the thick-fire test the depth of the coal on the grate varied from 8 to 20 inches, being heaviest at the rear end and lightest at the front end. On the thin-fire test the depth was maintained uniformly at about 6 inches. The coal used was New River semi-bituminous. The tests showed that the increased evaporation due to thin fires amounted to 11.5 per cent.

This result, which can probably be duplicated by tests of the same character made by other engineers, shows forcibly the importance of attention to boiler-room management. A

salesman who told a story of improving the economy of a whole boiler plant ten per cent. would probably be ranked as a gifted prevaricator. The change in conditions of firing effected such a result in this case, and "The Engineering Record" is firmly of the opinion that there are many plants in this country where the fuel bills could be cut down materially by careful attention to such matters. Owners of plants pay large sums for expensive equipment and then leave the most important part of the management, the firing of the coal, to the poorest paid and generally the least careful employés. The boiler room is the most vital part of a steam plant, and its management deserves careful, continuous supervision by somebody who understands the way to get the best service from the fuel.—"Engineering Record."

TURBINE BLADES.

Many experiments have been made into the wear of the blades of steam turbines to determine the cause of this trouble, and the latest to conduct an investigation are the firm of Sulzer Bros. It is undoubtedly a fact that the continued success of the turbine hinges largely upon this one point of blade wear. If the blades of a particular machine should have to be renewed frequently, because of deterioration and consequent loss in economy in the use of steam, that turbine would not make a commercial success. The Sulzer experiments indicate that the wear is due to the particles of water in the steam, and that if water is present in steam that flows at extremely high velocities, as is the case in single-stage turbines, the cutting is apt to be rapid. With superheated steam, or with saturated steam in multi-stage turbines, the cutting is not a serious matter.

The above experiments are in a measure confirmed, thinks an American contemporary, by the experience of those using steam injectors for boiler feeding. This cutting action seldom occurs in the steam nozzle, but frequently does take place in the combining tube where water is flowing at comparatively low velocities. It should be noted, however, that this water

is mainly the feed water pumped up by the injector, and contains more or less mineral matter, the extent of the cutting probably depending upon the quantity of such foreign particles entrained in the water. In the case of the turbine, however, the water particles suspended in the steam are or should be nothing but pure distilled water, formed by steam condensation after leaving the boiler. Probably no injector would give trouble by cutting if used to pump distilled water, assuming, of course, that the tubes were of a material that would not be corroded by the water. It is thought that no turbine, in which the steam is used at moderate velocity, will be likely to deteriorate rapidly under ordinary operative conditions.—"Page's Weekly."

A STUDY OF THE CORROSION OF CONDENSER TUBES.

By A. Humboldt Sexton.

(Reprinted through the courtesy of the "Engineering Magazine.")

The corrosion of condenser tubes is one of the difficulties which the marine engineer has constantly before his mind, for not only do the failures thus caused give him endless trouble and put him to considerable expense, but the corrosion takes place in so many ways and seems to be so erratic that it is almost impossible to guard against it, and in the minds of many engineers there is a feeling of uncertainty and insecurity which is far from pleasant.

I have had the opportunity of investigating certain cases of bad corrosion of condenser tubes, and have devoted a considerable amount of time to the experimental study of the subject; this paper is therefore submitted as a summary of the present position of our knowledge with reference to this subject, and at the same time as a contribution to its further study. It is not hoped to answer definitely all the questions that may be asked, or to solve all the difficulties that arise in practice, but perhaps to make clear the principles on which corrosion de-

pends, and to record results which may serve as a basis of future work.

At the outset it may be advisable to say a word or two as to the phenomena of corrosion as seen in condenser tubes, and all the points mentioned will be referred to later. It is assumed that the tubes are of brass, containing 70 per cent. of copper and 30 per cent. of zinc, which is the composition most generally used. When the tubes are removed from a condenser as being worn out, or at any rate as being unfit for further use, probably on account of the failure of some of them, they will usually be in one of three conditions:

(1) The tubes may be worn down nearly uniformly, the thickness of the metal being often considerably reduced, though, of course, it will be thinner in some places than in others. On making a section the remaining portion of the tube appears to be of brass quite unchanged.

(2) The tube may be considerably reduced in thickness, and, as before, to some extent uniformly; but there may be pitting here and there, and the inner surface of the tube seems to have become converted into copper—that is, it is dezincified—and this change in some cases may have gone to a very considerable depth.

(3) The tube may have worn down very irregularly, dezincification being well marked, often in patches, and sometimes extending at various places quite through the metal, giving the appearance of small plugs of copper, which may be either firmly adherent or quite loose; probably there will also be holes corroded through the tube—few or many, as the case may be. These plugs of copper and the holes will usually lie almost in one line, this line being at or near the bottom of the tube, when it is in position in the condenser.

Tubes in the first condition will usually have stood well, and may have been in use many years, those in the second condition may have lasted fairly well, but those in the third will usually have had a short life. The tubes which give trouble to the engineer are often found to be in the third condition.

In addition, there may be corrosion or dezincification to a

smaller extent from the outside or steam side of the tube, but this is usually due to causes different from those which cause the corrosion from the inside.

The tubes also frequently contain more or less deposit, but this varies so much in appearance that it is impossible to give a general description. Sometimes the tubes are nearly clean inside, and almost free from deposit; sometimes there is a thin skin of black material; whilst in other cases, and often where the tubes have stood badly, there is a heavy deposit, black or brown in color and containing a large amount of ferric hydrate. There are also usually patches, and sometimes a layer of a green deposit which is often very irregularly distributed, and sometimes clusters of white crystalline needles, or nodules of white crystals. Very frequently dezincification is very marked beneath the green deposit. The meaning of these appearances will be discussed later.

The corrosion with which we are concerned takes place in ocean-going steamships, in which the tubes are kept filled with sea water and so serve to condense the steam which surrounds them. The evidence is quite conclusive that sea water is the corroding agent, but its action may be accelerated or retarded by the presence of other materials and by the conditions under which the tube is placed.

Both copper and zinc are readily acted on by sea water under ordinary conditions and even at ordinary temperatures, and much more readily at higher temperatures. Brass, which is an alloy of copper and zinc, often with the addition of a small quantity of tin, is also acted on; but if of suitable composition, much less so than either of its constituents alone.

Sea water contains a considerable quantity of salts in solution, the one in largest quantity being sodium chloride, or common salt; it also contains, in solution, a considerable quantity of oxygen and of carbon dioxide. There is no doubt but that the oxygen in solution is largely responsible for the corrosion, and that the carbon dioxide plays some part in it; but the presence of this latter is by no means essential,

as corrosion will go on, though at a reduced rate, when carbon dioxide has been completely removed.

The formation of the deposit found in corroded brass tubes, in so far as it is derived from the corrosion of the tube itself, is due to the combined action of the water, salt, oxygen and, usually, carbon dioxide, as is evident from its composition. In one case, the deposit from a corroded tube was found to have the composition—

Copper oxide,	30.85
Copper chloride,	3.2
Zinc oxide,	30.8
Iron oxide,	11.4
Insoluble matter,	3.08
Organic matter, carbon dioxide, water,	20.67
Total,	100

These figures are of importance, for they show that in this case, at least, the deposit is basic chloride and carbonate of copper and zinc. It contains copper and zinc in nearly equal quantities, though in the tube the metals were present in the ratio 7.3, so that it is quite obvious, as indeed will be demonstrated later, that the zinc is more readily removed from the brass than the copper. It should also be noted that there is a considerable amount of oxide of iron (as hydrate), insoluble matter and organic matter, which could not have been derived from the tube itself, but must have been introduced from the outside. This is the case in every sample of deposit from badly corroded condenser tubes which I have examined.

A sample of the deposit formed by the action of clean sea water on copper was found to contain:

Copper carbonate,	29.77
Copper chloride,	8.39
Copper oxide,	59.38
Water,	2.46
	100

A large number of experiments have been made on the action of sea water on various metals, but more particularly on brass, with special reference to this question of corrosion.

Weighed pieces of metal have been exposed to the action of sea water, in some cases for many months, the water being frequently changed so that the action might not be interfered with by the removal of oxygen and carbon dioxide, which must have happened had the same quantity of sea water been used the whole time. Most of the experiments were made at the ordinary atmospheric temperature, but in some cases the tubes and samples were for some weeks heated to about 80 degrees centigrade during the day and allowed to cool at night. It would occupy too much space and would hardly be of interest to quote results of the experiments in detail, but a summary of the results obtained with brass may be of interest. Taking first of all a brass containing 70 per cent. of copper and 30 per cent. of zinc, which is the usual composition of the brass used for boiler and condenser tubes:

The action of sea water was very marked in every case, and, as might have been expected, was much more rapid when the water was heated. A green deposit was formed, exactly resembling that obtained from corroded condenser tubes. Sometimes this adhered to the metal, but at other times was quite loose. In some cases growths of white crystals of a zinc salt were noticed, exactly similar to those already mentioned as occurring in corroded condenser tubes. The zinc was always dissolved much more rapidly than the copper, but the proportion varies with circumstances. As a rule about five or six times as much zinc was dissolved as copper, whilst in the brass, of course, the ratio was 3.7. The action is more rapid at first; the green deposit which forms on the surface of the metal, especially if it be adherent, seems to some extent to protect it from further action.

It will be noticted that if the metals were attacked in this ratio in the tube from which was taken the deposit, the analysis of which has been given, a considerable quantity of zinc must have been removed in solution or in suspension.

When the brass contains 66 per cent. or less of copper, the action is very different; the amount of copper removed is small and the deposit is nearly white, consisting almost entirely of zinc salts. In a 66/34 brass tube the ratio of zinc to copper dissolved was 146.1 in one case and 214.1 in another, whilst with 50/50 brass, only a minute trace of copper was removed.

The actions by which a brass tube may be corroded are usually divided into three kinds:

(1) General corrosion, in which the copper and zinc are removed practically together (*i. e.*, in the ratio in which they exist in the alloy) by chemical or electric action, the tube thus diminishing in thickness, but otherwise being unchanged.

(2) Dezincification, in which the zinc is dissoved more rapidly than the copper, and thus a surface layer (or local spots of a material very rich in copper) is left.

(3) Erosion, in which the metal is worn away by the mechanical attrition of the water.

Corrosion of brass tubes always primarily takes the form of dezincification—except, of course, in the case of erosion—the sea water attacking the zinc more readily than the copper, the copper being left in a spongy form, which is very easily worn away. Whenever the copper and zinc seem to be removed together, *i. e.*, when general corrosion takes place, it is always because the spongy copper left by the dezincification has been removed by the mechanical action of the water aided by solid matter which may be in suspension, and this is why tubes which are worn away very uniformly show little sign of dezincification.

The question, however, remains to be answered: "Why is the action so much more rapid in some cases than in others? Why is it that whilst in some cases condenser tubes will last ten years or more, in others they fail in a few months, or occasionally even a few weeks?"

Obviously the fault—if fault there be, or at any rate the reason—must be in one of two places. It must either be due to something in the nature of the tubes themselves, or to the

conditions under which they have been worked. There is no alternative unless we assume some occult cause to explain the apparently erratic behavior. Each view has its advocates, the former being favored as a rule by engineers who use the tubes, but who are not familiar with the process of manufacture, whilst the latter is the view taken by the manufacturer.

I hold no brief for either side; I have investigated the matter as fully as I have been able, both in the laboratory and by practical examinations of cases of failure, and I am quite familiar with the methods by which the tubes are made and the processes through which they pass before reaching the engineer who will use them.

The tubes are, as already remarked, usually made of brass, containing 70 per cent. of copper and 30 per cent. of zinc, this being the mixture almost invariably specified. It is found to be as durable as any other, there being no advantage whatever to be gained by increasing the percentage of copper. A slight variation up or down has no deleterious influence, and, of course an increase in the quantity of copper is not likely to be injurious. Any careless work on the part of the founder is likely to lead to a greater loss of zinc, and, therefore, to a slightly higher percentage of copper. Many specifications are quite unnecessarily rigorous in the variations in the ratio of the two metals which they make allowable.

The composition of the tubes is always uniform. The composition at the two ends is always the same, and there are no spots, richer or poorer, in copper. This has been proved by many analyses; but quite apart from that, the conditions under which the tubes are made render such variation well nigh impossible, and the idea that there may be spots or patches in a tube differing in composition from the rest, which might set up electrolytic action and thus give rise to local corrosion, whilst believed by many engineers, is not in harmony with facts. The coppery plugs or spots already mentioned have been thought by some engineers to be due to fragments of unalloyed copper left in the mixture. This suggests an impossibility, for the spots are always round, and a fragment of

copper, if left unalloyed in the casting—which is, however, impossible—would be drawn out into a minute thread in the process of drawing through which the tube goes. These spots or plugs are of quite common occurrence in badly corroded tubes, though they are very often overlooked. The inside of the tube, when taken from the condenser, is usually covered with deposit; the outside is covered with grease; and it is not until these are cleaned away that the spots can be seen. The engineer, as a rule, is not concerned to make a minute examination of the failed tubes; he is anxious to get them away and to get new ones in their place, and hence, unless there is some special question likely to arise, the tubes will not be examined and the spots will not be detected. These spots of copper are certainly due to dezincification, as also is the layer of copper sometimes found inside the tube, and the copper almost always retains a certain amount of zinc. In one case a fragment of a coppery patch from a corroded tube contained 3.7 per cent. of zinc, showing that most of the zinc from the original brass had been dissolved away at that spot, and a specimen of dezincified layer from inside another tube contained 3.84 per cent. of zinc.

It is very doubtful whether the use of specially pure metals is any great advantage, though it is no disadvantage, and it is wisest to err—if at all—on the safe side; but the use of specially pure and, therefore, costly metals, seems to entail increased cost without any corresponding advantage. Nothing in the chemical composition of the brass, if it be approximately 70 and 30, will account for the variations in the durability of the tubes. Indeed, it not infrequently happens that with tubes as far as can be detected in all respects indentical as to composition and make, one set will wear well, whilst another will corrode with extreme rapidity.

The mechanical structure of the metal has sometimes been held to be responsible for corrosion. Brass containing 70 per cent. copper and 30 per cent. zinc is a homogeneous alloy. On cooling, it crystallizes as a whole and does not separate into two constituents as the poorer brasses do. It behaves in

this respect as if it were a chemical compound, but that it is not a true compound is shown by the fact that the elements are not present in simple atomic ratios, that the composition may be varied within wide limits upwards, and to a small ex-

Fig. 1.—Microsection of Brass Condenser Tube, Showing Coarse Crystallization. (Magnified About Fifty Diameters.)

tent downwards, without altering the structure, and by many of its chemical reactions. It crystallizes as a whole and forms a homogeneous crystalline mass. In the casting the crystals are very large and distinct, but they are broken up during the drawing, and are reformed when the metal is annealed. The actual size of the crystals in condenser tubes varies very much, the difference being probably due to variations in the

time and temperature of annealing. Figures 1, 2 and 3 are microsections of brass condenser tubes magnified about fifty diameters. It will be seen that the size of the crystals is very different in the three cases. This difference in the size of the crystals may have a profound influence on the physical properties of the metal, but has no influence whatever on its resistance to corrosion.

It has been suggested that the crystals being differently oriented might set up galvanic action, so that the lines of junction of the crystals might become centers of corrosion. There is no evidence whatever in support of this idea; and

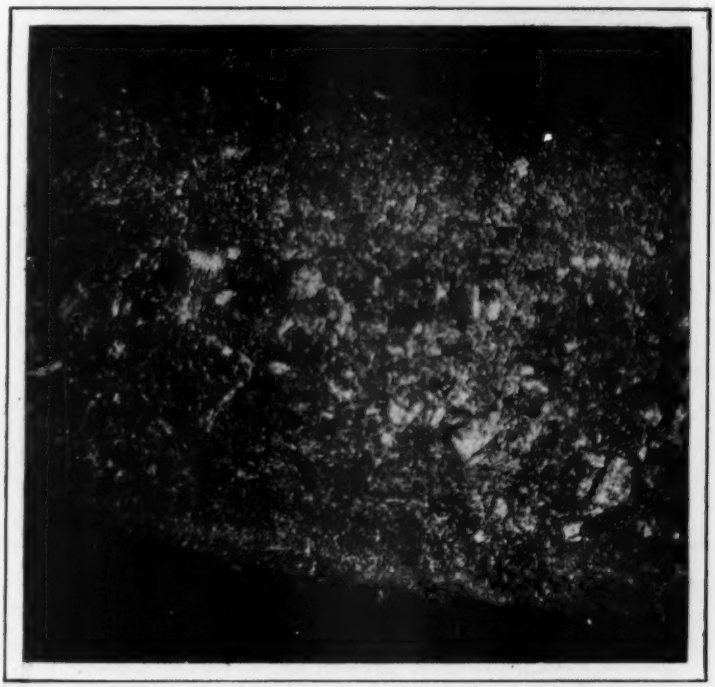


Fig. 2.—Microsection of Brass Condenser Tube, Showing Medium and Rather Irregular Crystallization. (Magnified About Fifty Diameters.)

even if it were true, it would not account for the formation of pits and corrosion patches a considerable distance apart. I have found that dezincification bears no relation whatever to crystal structure, the line of action as often passing right across a crystal as round its margin.

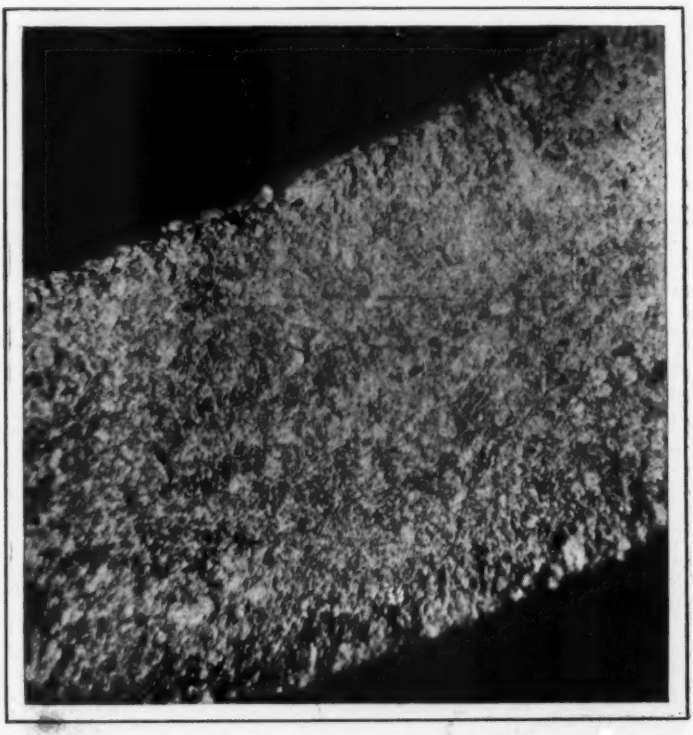


Fig. 3.—Microsection of Finely Crystallized Brass Condenser Tube. (Fifty Diameters.)

It has been contended also that the larger the crystals the more rapid will be the corrosion. This, however, is certainly not the case. I have found that the size of the crystals has no influence whatever on the corrosion of the metal. A very

large number of corroded tubes have been examined, and it has been found that good and bad tubes—the quality being measured by the amount of corrosion that has taken place—are impartially distributed between those with a coarse, medium and fine structure.

Among methods that have been suggested for increasing the durability of condenser tubes may be mentioned the addition of a small quantity of tin to the alloy. The British Admiralty, for instance, uses an alloy which contains 70 per cent. of copper, 29 per cent. of zinc and 1 per cent. of tin; but it is very uncertain whether this is any material advantage. The coating of the tubes with tin has also been tried, but the protective action, if any, is very small. I am convinced that for this particular purpose nothing is better than 70/30 brass.

I feel quite certain that the cause of variation in the durability of condenser tubes is not to be found in the chemical composition or physical structure of the metal, nor in any variation in the process of manufacture, nor in anything connected with the tubes; indeed, the tube maker, while keeping to the specified composition and passing the tubes through the usual tests for soundness, could not, if he tried, turn out a tube specially liable to corrosion. This is, of course, not the usual opinion of engineers. They say: "Here are two steamers working under exactly similar conditions, and whilst in one the tubes have stood well, in the other they have corroded very rapidly; therefore, the reason must be in the quality of of the tubes." This dilemma may, however, be put in another way. Here are two steamers fitted with exactly similar tubes, selected haphazard out of one large parcel; in the one steamer the tubes have stood well, whilst in the other they corroded rapidly; therefore there must be a difference in the condition of working. The latter is certainly the correct view, for there are so many possible variations in the conditions of working that it is impossible to decide when these are uniform. Two steamers moored at the same berth on a river may be under very different conditions, and so in many other ways there may be differences of condition which, together with differ-

ences in the way in which the condenser is treated by the engineers in charge, may account for the differences in the corrosion of the tubes.

The solvent action of sea water will account for the slow and steady dezincification which will ultimately wear away the tube; but this will take years, and something more is necessary to explain the irregular corrosion which is so destructive. The corrosion into holes in particular spots seems to indicate some local cause, and at once suggests electrolytic action, the tube itself being the anode; but if this is the case, the difficult thing is to find what can be cathode. It must obviously be some substance strongly electro-negative to the brass.

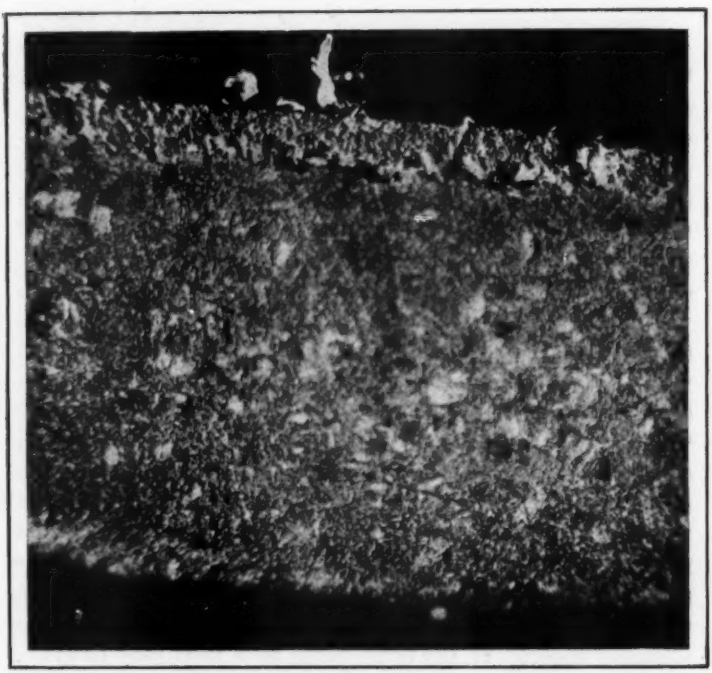


Fig. 4.—Condenser Tube, Showing Layer of Dezincification. (Fifty Diameters.)

Many experiments have been made as to the influence of the contact of various substances with 70/30 brass—pieces of condenser tubes being used—immersed in salt water, and though the results obtained are not new, they tend to throw some light on the subject under discussion.

Malleable iron and zinc in contact with the brass, as might be expected, completely protect it from corrosion, being strongly electro-positive to it; on the other hand, when a fragment of graphite was placed on the metal, a well marked spot of dezincification (*i. e.*, copper spot) was formed beneath it. The spot was, of course, only superficial, the length of time the experiment was continued not being sufficient to allow the action to go on to any depth. With a piece of coke an exactly similar result was obtained. In both cases the action was more marked when the water was heated. A layer of blacking, which is nearly pure carbon, increased the action of sea water, and also increased the ratio of zinc to copper dissolved. Similar results were obtained with other substances. The green oxychloride of copper seems to protect the copper, and thus to increase the ratio of the zinc dissolved. Ferric hydrate is stated by Professor Tilden to cause rapid corrosion. With a fragment of iron scale, dezincification under the scale was well marked, but copper scale has no such action.

Foul river water has been accused of causing rapid corrosion, and it may be admitted at once that there are many cases of rapid corrosion in the case of steamers plying alternately on such waters and on the sea—perhaps more than on steamers plying only on the deep sea; on the other hand, in many such steamers the tubes last quite the normal time. It will, of course, be admitted that acid waters will be likely to increase corrosion considerably, though 1 per cent. sulphuric acid has little more action than sea water; in any case, the action of an acid water would be expected to be nearly uniform, and, whilst it might accelerate corrosion, would hardly be likely to cause pitting. It is only in very exceptional circumstances, as for instance when acid refuse is discharged into a river, that the water will be sensibly acid.

Brass tubes exposed for four months to Clyde water—and the Clyde is not regarded as the purest of rivers—taken from the Broomielaw, did not lose seriously in weight, but only became tarnished by the formation of a dark layer, apparently of copper sulphide. Some samples of brass tube were moistened with ammonium sulphide so as to produce dark stains of copper sulphide, and were then put into sea water. The action was at once very marked, the ratio of zinc to copper dissolved being increased to 16.1 per cent. in one case and 12.1 per cent. in another. It would seem, therefore, that a river water containing sulphides might cause a deposit of sulphide which on the tubes being filled again with sea water would set up corrosion, and this being repeated day after day vigorous corrosion might result.

Other suggestions have been made, as, for instance, that the corrosion may be due to electric currents, from the electric lighting or other electric plant on board ship, especially where the single-wire system is used. This may be an occasional cause of corrosion, but I do not think it is at all frequent. It is difficult to see how stray currents can find their way into the condenser, so as to cause electrolytic action. In a paper read before the Institution of Naval Architects on April 3, 1903, Mr. A. W. Stewart describes the corrosion of copper bilge suction pipes on certain steamers of the Russian volunteer fleet, and he comes to the conclusion that in this case electric currents had nothing to do with the corrosion. Corrosion does not seem to be more common in steamers with electric fittings than others, nor does it seem to have increased since the use of electricity became common.

In the above mentioned case, Mr. Stewart ascribes corrosion to the presence of oils and grease. But it is obvious that this is not likely to be the case inside condenser tubes, where grease or oil could not ordinarily find its way. Corrosion on the outside of condenser tubes, which is usually on a small scale, is no doubt often produced by oil or grease carried over by the steam.

I have come to the conclusion that rapid and irregular cor-

rosion, as distinguished from that due to the normal action of sea water, is almost invariably due to electrolytic action set up by the contact of particles of substances electro-negative to the brass, probably in most cases carbon.

Where can these particles come from? Frequently they may come from outside, especially if the water be drawn from such a position that cinders thrown overboard may be drawn in with it; and cinders have actually been found in corroded condenser tubes. In some cases particles may be derived from the cast-iron doors of the condenser itself, if these are not well protected. A fragment of cast iron would be rapidly oxidized, and for a time would act as a protective agent, but as soon as the iron was dissolved the particles of the residual spongy mass, consisting mostly of graphite, would be electro-negative to the brass and would cause rapid corrosion. If the iron of the door rusted, the graphitic mass would be easily detached, and might be carried into the tube. As already remarked, the corrosion spots usually lie in a line along the bottom of the tube, this indicating that they have been caused by something which settles there. It is also noteworthy that the deposit in corroded tubes nearly always contains a considerable quantity of oxide of iron, which must have been derived from some external source. When a steamer plies on a foul river, the conditions are much more likely to lead to corrosion than on the open sea. On such a river as the Clyde, cinders and fragments of charcoal frequently float on the surface, and may readily be drawn into the condenser. The water may contain sulphides, as evidenced by the very strong odor of sulphuretted hydrogen that may often be detected, and thus patches of sulphide may be formed which may subsequently set up action. Sticky organic and oily matter may frequently be present, and these, though they may have no direct action, may cause the adherence of particles which may in their turn lead to corrosion, especially when coming in contact with sea water. Indeed, the composition of foul river water is so complex and variable that it is difficult to say what action may not take place. The conditions also of such a river vary much with

the seasons and other circumstances. In time of drought, when the water is low, it will be more strongly charged with impurities than in times of rain, when there is abundance of water. The discharge of acids and other impurities from works is not continuous, but takes place at intervals only, and there may be variations even from hour to hour; and even if the conditions are the same, the state of the tubes as to deposit, and other conditions, may much modify the action.

As to the cure for irregular corrosion, there is none—at any rate after it has made progress; but, like many diseases, if it cannot be cured it can be prevented; and I am strongly of opinion that it is always preventable.

1. The corrosion from the presence of solid particles can take place only if such particles are allowed to rest in the tubes. If the current be strong, therefore, corrosion is little likely to take place, while if it be sluggish, corrosion is very probable. Should a tube become partially stopped for any reason, that tube is specially liable to corrosion. Sluggish circulation is a very common cause of corrosion.

2. The tubes must be frequently cleaned, so that any deposit which is forming may be removed. This is of special importance in steamers running in foul rivers, which may readily pick up substances which may cause the adhesion of objectionable material.

As has been pointed out, tubes that have corroded badly are almost always characterized by the presence of a heavy deposit.

3. The tubes should never be left full of water when the steamer is at rest, but should be run dry and perfectly washed out with clean water, as soon as the day's work is done. This, too, is of special importance in steamers running on foul rivers where objectionable matter may be drawn in, which, during the period of rest, will settle to the botom of the tube and form a lodgment from which it will not be displaced when work is resumed, and so corrosion may be set up, and once started it will go on rapidly under the deposit formed.

In conclusion, I am quite convinced that if the condensers

be well designed and these precautions be taken, very little will be heard of unduly rapid corrosion; but that unless such precautions are taken the tubes will always be liable to erratic and rapid decay, and the conditions being entirely beyond any power of calculation, no estimate or guarantee can be given as to the probable life of a tube.

Some of the experiments cited are only tentative, and perhaps some of the conclusions may not be fully proved; but I feel quite sure that on the whole they are sound and in the right direction. I will be much obliged if engineers and others who come across cases of corrosion which either bear out or tell against these conclusions will communicate with me and send samples of the corroded tube deposits and anything else that may help to complete the investigation of the causes of the corrosion of condenser tubes and other under-water fittings on board ship.

THE "CONTRAFLO" CONDENSER.

The following notes on a very interesting type of condenser are taken from a description of Messrs. Richardsons, Westgarth & Co.'s Steam Turbine Works, Hartlepool, which appeared in "London Engineering":

The condenser is a very interesting part of the plant, and has been given the name of "Contraflo." This condenser is illustrated in Figs. 1 to 3. The particular condenser here illustrated was constructed for the Hammersmith Corporation Electricity Works, and is for a 1,500-kilowatt steam turbine. The exceptional power of the steam turbine to take advantage of high vacua has led Messrs. Richardsons, Westgarth & Co. to consider how best to attain that end, with the result that they have adopted this type of condenser in their turbine department. The results, they report, are certainly remarkable; but as the efficiency of the apparatus is being exhaustively tested by Professor Weighton at the Durham College, Newcastle-on-Tyne, and as the results will be communicated in a paper to be read by the Professor before one of the engineering institutions, it will suffice at present if we give only an outline

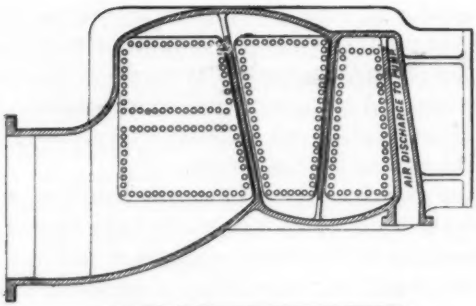


Fig. 3.

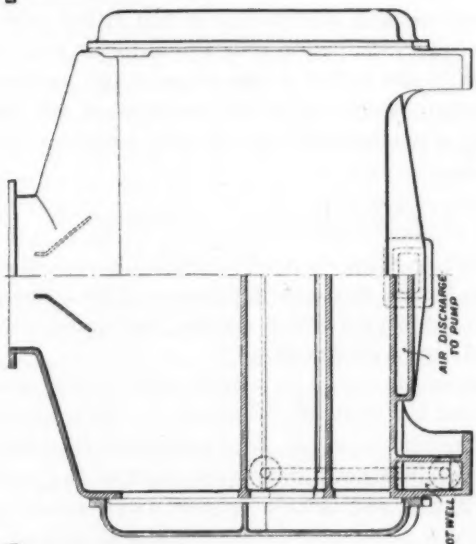


Fig. 2.

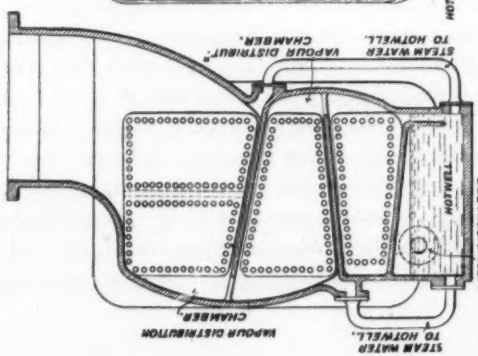


Fig. 1.

description of the apparatus. It may be added that Messrs. Richardsons, Westgarth & Co. have presented a "Contraflo" condenser, with the necessary plant for its working, to the college, where it will be available for research purposes.

It will be seen by the diagrammatic sectional views, Figs. 1, 2 and 3, that the condenser is divided into communicating compartments of decreasing capacity; and steam and water are drawn off simultaneously from each compartment. Fig. 1 is a cross section through the hotwell end, and Fig. 3 is a cross section through the air-discharge passage. Fig. 2 is a front elevation, half in section. The exhaust steam passes in at the top, flowing through the nests of tubes to the vapor distribution chamber—shown on the left of Fig. 1—common to the two top compartments. The vapor uncondensed passes through the orifice shown in the top diaphragm, dividing the upper two compartments. The water resulting from the condensation that takes place in the upper compartment flows down to the right of the top compartment, owing to the inclination of the dividing diaphragm, and from thence passes through the pipe shown to the hotwell at the bottom of the apparatus. The hotwell is of limited capacity, and the communicating pipes are of sufficiently small dimensions. The vapor uncondensed in the top compartment flows from the lower part of the left-hand vapor distributing chamber (Fig. 1) amongst the tubes in the middle compartment, and a further quantity is there condensed. The water thus produced flows through the pipe on the left of the compartment to the hotwell, any vapor still remaining passing to the upper part of the lower vapor distributing chamber, on the right in Fig. 1, and from thence through the orifice shown, to the lower part of the latter chamber, and finally to amongst the third or lowest nest of tubes, where the condensation is completed, the resultant water flowing directly to the hotwell.

By the gradual elimination of the vapor in the earlier compartments, the air, when it reaches the bottom of the condenser, is comparatively dry, and being drawn by the air pump through a port extending throughout the length of

the tubes (see Figs. 2 and 3) uniformity of flow is maintained over the entire surface, a temperature resulting which is considerably below that of the water in the hotwell. The combined effect of the rapid trapping and removal of the condensed steam (or steam water), and of the low temperature to which the air is reduced, results in the temperature of the steam water being often well above the temperature corresponding to the vacuum produced. This might be anticipated from the fact that a large part of the water of condensation is drawn immediately to the hotwell from the upper and hotter zones of the condenser, instead of all being carried through to the region of lowest temperature. The important effect obtained is an abnormally high temperature of feed water, and consequently a distinct gain in thermal efficiency. Other practical results are a diminution in the amount of circulating water, and a considerable reduction in the size and weight of the apparatus. The design represents an important advance in surface condensers, and will be of special interest in relation to the high vacuum requirements of steam turbines; whilst decreased weight and space occupied will strongly appeal to the designers of warship machinery. The tests being carried out by Professor Weighton are of a very exhaustive character, and will be of great technical interest.

THE FUTURE OF THE CRUISER.

Considerable significance attaches to the statement that the *Natal*, launched last Saturday, may be among the last armored "cruisers" floated from private works for the British Navy; there are three of the class still building in the dockyards. It is suggested that in future this type of vessel will be merged with the battleship, because of the great powers of offense and defense required in all large ships of war. This general principle, without any details, and obviously without reference to anything but widely entertained naval opinion, was expressed by a high authority on naval construction—Mr. Albert Vickers—who, it will be remembered, also foretold the

great increase in the number of high-power guns in battleships, since decided upon in all projected ships, to the exclusion of weapons of intermediate caliber. Our new ship, the *Dreadnought*—the keel of which was officially "laid" this week—will have ten 12-inch guns, the other pieces being only those for defense against torpedo-boat attack. The United States naval authorities have deferred further progress with the design of their new ships until the legislature meets to give permission to increase the displacement in order that a similar arrangement may be adopted. France has decided to enormously augment the gun power of her new battleships, and Germany is seriously considering the question of increasing the depth of water in canals and harbors, so that larger ships may be built to accommodate the modern equipment of guns. The same reasons for increase in ordnance are applicable to armored cruisers; but here there are complications in the problems of design which do not make it so easy to realize the ideal conditions.

The trend for the past two or three years has been towards an increase in the power of the ordnance of warships, and a very considerable impulse has been given to opinion in this direction by the recent naval operations in the Far East. There, under normal conditions, and at long range, the gun failed to penetrate modern cemented thick armor. We are in possession of important details regarding the destruction done to the Russian ships at Port Arthur, since salved by the Japanese Government, and hope soon to have authority to publish these; but it may be said generally that the damage was surprisingly small. In only one spot was the main armor perforated in any of the ships; in both the *Peresviet* and the *Retvisan* there were four holes, and in the *Poltava* one hole, under the armor belt. Few of the shots which perforated the upper deck passed through the armored deck. In the *Peresviet*, which suffered most, four shots out of twelve found their way below the protective deck. These perforations seem to have been largely due to the 11-inch and other guns mounted by the Japanese forces on 203-Meter Hill and other positions around

Port Arthur, firing at a range of about four miles and at a high angle, so that the projectiles plunged on to and penetrated the decks of the ships, although these weapons were of less power than naval guns of the same caliber. Moreover, Admiral Togo in his actions against the Port Arthur squadron, found it desirable, in order to conserve his relatively small battleship fleet, to fight at long range. It was, no doubt, the consciousness of the ineffective fire at such range—as demonstrated in the Gulf of Pechili—that induced him, when he came to the crucial battle of the Sea of Japan, to refuse to waste ammunition until he got within 6,000 meters, notwithstanding that the Russians opened fire at 9,000 to 10,000, meters. In other words, it was clear to him that even the 12-inch gun is only suited for engagements at about 3 to 3½ miles range. The need, as we have shown, of superior penetrating power of projectiles has been recognized in the case of battleships; and it is equally obvious that if cruisers are to be brought into the line of battle to fight at such range, guns of 6-inch or even 7.5-inch caliber are totally inadequate, and their armor protection may be insufficient. In the subsidiary, or purely cruiser, action in the Sea of Japan the ineffective attack of the 6-inch gun was further established; so that, according to present experience, it seems that the 9.2-inch gun is the absolute minimum for primary armament for large ships. Indeed, there are authorities who consider that the 10-inch or 12-inch breechloader is the best weapon, notwithstanding that its rate of fire is unavoidably less than that of the 9.2-inch piece. But power in each separate shot is the one thing needful.

The question next to be determined is as to whether armored cruisers should be included in the line of battle, or, rather, whether they have a distinctive function. Admiral Togo utilized them in that position, but this was probably of necessity rather than by choice, owing to his deficiency in battleships. There is, however, the significant fact that the practice of our own admirals during the maneuvers of recent years has been to similarly use them. It is felt that without guns of 9.2-inch or higher caliber no warship—as distinct from torpedo craft—

is of general utility. And this practically leaves no alternative to the naval constructor but to provide so-called cruisers with guns of a caliber almost equal to those of battleships. If the cruiser is not to become abnormally large, the question is as to the number of weapons permissible. Is the cruiser of the future to have high speed at the expense of protection and the number of large-caliber guns? If she is to fight in the line, is heavy armor not almost as essential as powerful guns? But before considering these questions it is necessary to inquire as to whether there are not functions, as, for instance, in the protection of commerce, which demand a higher speed than is essential in a battleship.

This inquiry as to the future of the cruiser raises the whole question of the strategy of the British fleet. We are primarily concerned with our own Navy—an explanation here called for, since strategy and tactics must be based on the relative strength of combatants. A weak power cannot afford to—certainly need not—accept the main strategy of such a strong fleet as that possessed by Britain. It has always been regarded as a first principle that the enemy's coast must be the objective point of our fighting squadron, and that the maintenance of our sea power must be the ultimate aim. So long as our fleet is strong enough to take the offensive against the enemy's ships the command of the sea cannot be said to be in peril. The aim of our fleet, therefore, would be to engage the enemy in squadrons or to contain his ships in their own ports. In such case the cruiser would take her place in the line in fleet actions. It has been popularly regarded as essential that we should have cruisers for protecting our commerce against fast hostile cruisers, but the official view of the present Board of Admiralty is that "the attack or defense of commerce is best effected by concentration of force, and that a dispersion of strength for either of these objects is the strategy of the weak, and cannot materially influence the ultimate results of the war." This view is also taken by Captain Mahan, who, in his "Influence of Sea Power Upon History," says that the harassment and interference with commerce and the at-

tempt to stop commerce is, when regarded as a primary and fundamental measure, "a most dangerous delusion when presented in the fascinating garb of cheapness to the representatives of the people; especially is it misleading when the nation against whom it has to be directed possesses, as Great Britain did and does, the two requisites of a strong sea power—a widespread, healthy commerce and a powerful navy."

The main strategy laid down by the Admiralty thus involves a concentration of forces; and while our naval authorities do not say that there will not be harassment of our commerce, they refuse to recognize this as a dangerous factor, and do not anticipate that it will necessitate a departure from the main function of our fleet. The percentage of loss in captures by the enemy will be small, partly because our trade routes need not be defined. Even should the enemy's force be dispersed to harass commerce, it will be possible by fleet tactics to counteract the moral influence of such captures, if not to involve the recall of the commerce destroyers. In the last resort it will be possible to detach a superior number of vessels to run the commerce destroyers of the enemy down. The long-entertained idea, therefore, that commerce-protecting cruisers are necessary has been departed from, if it ever found full acceptance, and in any case it need not materially influence the design of vessels which are almost primarily regarded as for fleet engagements. Certainly the opinion is growing that, as Mr. Vickers put it, the type will be merged into the battleship class. This view is entertained not only by many strategists and officials in this country, but also by naval experts in France, America, and even in Germany. But, while it is conceded that the armament of the cruiser should be brought more into line with that of the battleship, there is not the same general agreement regarding the speed and the armor protection of the high-speed battleship which may be "detached," as a last alternative, as the Admiralty put it, to run down the commerce destroyer of the enemy.

In arriving at the compromise between gun, armor and speed, the limiting feature is size, which includes cost. Speed

is the most costly element of design, for the temptation of the naval designer is to increase the length of hull, as this is of itself conducive to propulsive efficiency. This addition involves a larger target to be protected, and therefore a greater addition to the price of the ship than the augmentation of any other dimension. These problems of compromise are also exercising the minds of naval critics in France, owing to the determination to alter the type of ships to be built, in consequence of the experience in the Far East. But in their case there are factors not obtaining here. It is no disparagement to our friends across the Straits to say that circumstances prevent them spending so much upon their Navy as is possible with our Government, and consequently it is necessary to minimize, as far as is prudent, the expenditure on any one ship. We, on the other hand, are fortunate in being able to place finance in an entirely subsidiary, although nevertheless still an important, position. While, therefore, our Admiralty—and especially the present Board—are certain to give the greatest value for a given expenditure, they do not experience the same monetary limitations as other naval authorities.

Again, the tendency of the weaker power is to depend upon speed, so that in the event of a superior force being encountered, there is the choice of combat and of escape. But no war can be determined without a decisive engagement, and our maintenance of the command of the sea necessitates that our enemy's squadron shall not escape to exercise the discomfiting functions of a "fleet in being." It is therefore easy to understand why many French critics have a leaning towards increasing the speed at the expense of the number of high-caliber guns fitted to the battleship, or the cruiser battleship; while we must still excel in speed as well as in the powers of offense and defense.

Speed was a most important factor in Admiral Togo's success. It is true that his best speed during the battle of the Sea of Japan was 15 knots, but this was quite sufficient to outmaneuver the Russian fleet with their maximum of 12 knots.

As a consequence, Admiral Togo succeeded, time and again, in heading off the Russian columns, and each time he crossed the head of the lines he so concentrated the fire of all his ships on the leading vessel of each Russian line as to fatally damage, first, the *Osliabia* and the *Souvaroff*, and later the *Alexander III*, which had become leader of the column. Several contributions have recently appeared in the "Moniteur de la Flotte," one of the foremost of the French naval papers, from writers of practical experience and professional ability. They consider the question from the standpoint of the French fleet, and many of them take the view that the future cruiser battleship must sacrifice a few of her large guns in order to increase the speed; but, for reasons of size and cost, they cannot attain to the rate of steaming of present-day cruisers with inferior armament.

M. C. Pierreval makes a strong point, especially from the point of view of the French fleet, when he indicates that high speeds involve greater risks of breakdown, although these may be much minimized by good design and workmanship and by a sufficiently large and experienced staff of engineers. He assumes the meeting of two squadrons, one of 18 knots and the other of 21 knots, differing only by the predominance of speed obtained at the expense of gunpower. In the probable event of the fastest squadron having to run away, he points to the grave risk of a machinery accident. This would involve the sacrifice of the disabled ship; otherwise, the squadron, standing by its "lame duck," would be compelled to fight, notwithstanding inferior guns. He therefore contends that as the gain in speed is more or less uncertain, the preference should be given to the gun. It is computed by one French writer that in a large cruiser an increase in speed of 3 miles per hour would involve an addition of 10,000 horsepower. This, at the minimum computation, would mean 700 tons more weight, which seems rather low. It is equal to 10 per cent. of the defensive power, or to a turret with two 12-inch guns, *i. e.*, 16 per cent. of the offensive power. The problem

is to determine which of these sacrifices in the fighting qualities is to be made.

In the case of the British fleet we are, fortunately, not so limited in respect of the size of ships as in most other countries, either by the draught available in harbors, by the capacity of our docks, or by the length of our purse. In our new battleship, the *Dreadnought*, we have been able, through the skill of our designers at the Admiralty, to meet all of these requirements. As has been already indicated from semi-official sources, the gun power has not only been enormously increased to ten guns of 12-inch caliber, but in order to add the required increment to the speed, machinery has been adopted which will probably involve an increase of about 500 tons in weight, and, at the same time, the armor protection has been improved. In our new high-speed battleships—the name which must be applied to the armored cruisers of the future—a corresponding general advance will doubtless be realized.

The question is the degree of the advance in each quality. Can the *Dreadnought's* armament be adopted in combination with a higher speed? Two miles per hour more would certainly add to the weight of machinery an equivalent to that involved in at least two 12-inch guns with barbette and mountings. The difference might be met by installing 10-inch guns of 30,000 foot-tons muzzle energy, instead of the 47,000 foot-tons of the 12-inch weapon, or by a combination of 12-inch and 9.2-inch guns. There should be an increase in the speed of our next cruiser battleships. A reduction will, in all probability, be required to be made in the number or caliber of high-power guns to be carried, as compared to the *Dreadnought;* but it is certain that these fast battleships, while equalling the speed of armored cruisers, will be of superior gun power to the existing line-of-battleships. As to whether the changes will be made in connection with the armored ships to be laid down under this year's program remains to be seen.—"Engineering," October 6, 1905.

THE CLASSIFICATION AND PECULIARITIES OF STEAM TURBINES.

Although the information lately published concerning steam turbines is voluminous, it is difficult to find any unbiased general statements concerning all types of such machine. One reason for this is the fact that few people agree even on such a general subject as the classification of steam turbines, and in view of the different views on this topic it is only natural that equally diverse opinions should be held concerning details. Professor Storm Bull recently read a paper before the Western Society of Engineers in which the classification and peculiarities of turbines were discussed at some length, and this paper deserves attention because the author advocates the adoption of certain German views concerning classification which are more rational than those customarily held in the United States.

The first steam turbine, the 10-H.P. wheel built by C. A. Parsons in 1884, was a reaction machine, while the next type, the de Laval wheel of 1889, was an impulse wheel, using the terms commonly employed. The early machines were at the extremes of turbine possibilities, and development has taken place along the whole line of types between them, which should be classified, Professor Bull says, according to the manner in which the steam acts rather than by their external appearance or speed. The only essential difference between water and steam turbines is that in one the fluid is incompressible while in the other it is compressible. Unfortunately, turbine water wheels are all really reaction wheels, although where all the energy of the water is developed as a jet the wheels are commonly called impulse wheels. This rather unfortunate use of terms has led to the introduction in Germany of new names for these classes, velocity and pressure turbines, which Professor Bull advocates for general adoption. In a velocity turbine, like the de Laval, the steam expands exclusively in passing through the guide wheel or nozzle, and no expansion takes place in the turbine wheel proper, so the pressure is exactly the same on entering the turbine wheel as on leav-

ing it. This is only possible if the guide and turbine wheels are in separate chambers. In the pressure type, like the Parsons turbine, the pressure gradually decreases from one end of the machine to the other, and the steam expands in the turbine wheels. The classification of steam turbines is not so simple as that of water turbines for another reason, the practicability when using steam of introducing multistage operation, a subject Professor Bull discusses at some length.

With steam of even fairly moderate pressure, the velocity of efflux into vacuum is several thousand feet per second. According to both theoretical and practical hydraulics, the velocity of rotation of the moving blades or buckets should be not far from one-half of that of the steam, in order to get the best efficiency. For steam of 150 pounds pressure by the gauge and a vacuum of 27 inches, the velocity of the steam is not far from 4,000 feet per second; consequently, the velocity of rotation should be nearly 2,000 feet per second. De Laval lets his smaller size turbines run with 30,000 revolutions per minute, and in that manner obtains a velocity of rotation which is not too far away from what it ought to be. For turbines of greater power, the maximum capacity being 300 horsepower, he found it necessary to use a larger wheel, and the limit of the size is determined by the number of revolutions and the strength of the material of which it is made. With these larger sizes the number of revolutions had to be cut down to some 15,000, even though they are made of the very strongest nickel-steel, and the velocity of rotation is only about two-thirds of what it ought to be for the very highest efficiency of the turbine. It has been found impossible to go higher than 300 horsepower, notwithstanding the most careful study of the problem, and de Laval had to invent the flexible and self-adjusting shaft before his turbine could become a practical success.

The purpose of the many-stage or multiple turbine is to get around this difficulty with respect to the high velocity of efflux of steam. Instead of changing the energy of the steam all at once into kinetic energy in a nozzle like that of de Laval, this may be done in several steps, the energy corres-

ponding to each step being utilized or absorbed by the wheel before the next step is reached. If, for instance, there are fifty steps—as in many of the Parsons turbines—and the difference between the pressure in the boiler and condenser be 150 pounds, then the average drop for each step is only 3 pounds, and the corresponding velocity of the steam comparatively small. However, the impression must not be left that in a many-stage turbine the pressure drop is constant for all the wheels. It may be so, but ordinarily it is only approximately so, and it is not essential to a high efficiency. By this method, then, it is possible to have a very much smaller velocity of the steam, and consequently an equally smaller velocity of rotation, so that the troublesome problem of finding a material which could withstand the centrifugal force generated by the high rotation speed has been gotten rid of.

Returning to the classificution of steam turbines, the question is whether these many-stage turbines are pressure or velocity turbines. The answer is that there are many of both kinds. In a many-stage velocity turbine the pressure in the smaller clearance between the guide wheel and the turbine wheel should be the same as between this latter and the next following guide wheel, and the drop of pressure takes place in the guide wheel. Practically there will be a very small drop of pressure in the turbine wheel, just sufficient to overcome the frictional resistance which the steam has to overcome in passing through the wheel, even though, as is now well established, the drop of pressure does not represent a total loss to the turbine; the real loss being only the lowering of the energy from the pressure on entering the guide wheel to that in the turbine wheel. The best known velocity turbine in the United States is Curtis', in which the steam flows from a series of nozzles into the first turbine wheel, from which it flows through a guide wheel into a second turbine wheel, the pressure remaining nearly constant from the entrance to the first until the exit from the second turbine wheel. From this latter the steam collects in a small reservoir, and is then expanded through nozzles and flows through a turbine wheel, again

through a guide wheel, to a second turbine wheel and into a second reservoir, and so forth. In this manner there may be three or four pressure stages, each one subdivided in two velocity stages, this latter subdivision serving also to reduce the velocity of rotation.

The Rateau turbine belongs also to the velocity class of turbines. In this there are as many pressure stages as there are turbine wheels, and the space between each turbine wheel and the next following guide wheel forms a reservoir from which the steam flows into the guide wheel, in passing which the steam expands and changes part of its heat into kinetic energy. That this turbine must be a velocity turbine seems evident from the fact that the guide wheels are not provided with openings all the way around, the turbines being what are called "partial wheels." As the steam expands, after passing through successive wheels, the number of the openings or buckets in the guide wheels increases until it becomes necessary to either widen these openings radially or to enlarge the wheels themselves. The important point here is that the expansion of the steam takes place in the guide wheels, except so far as it is necessary to overcome the resistance in the turbine wheels; hence the Rateau turbine is a velocity turbine, but of many pressure stages.

In many-stage pressure turbines the pressure is gradually reduced from the moment the steam enters the first guide wheel until it enters the condenser. The expansion of the steam takes place both in the guide and turbine wheels, the velocity of the steam also increasing, and more rapidly, toward the end than to begin with. The most prominent example of the class of steam turbines is that of Parsons. Depending on the pressure of the steam and of the power of the turbine, the number of wheels in a Parsons turbine may be as large as eighty, although about sixty probably is more nearly the average. It is, of course, quite possible to build a Parsons turbine with a very much smaller number of wheels, and evidently such a turbine would be very much cheaper to build. The drop of pressure would then be correspondingly greater

for each wheel, also the velocity of the steam as well as that of rotation, and consequently the number of revolutions. If, therefore, it be desired to design a wheel with a comparable low number of revolutions, it is necessary to use a large number of wheels, and presumably this consideration has been the leading one at least in the choice of the number of wheels in the Parsons wheel as it is built. Questions of the strength of materials necessarily also enter the problem, and to a certain extent also the efficiency of the turbine. With a high velocity of the steam the friction per running foot necessarily must be greater than for a low velocity, as it is supposed to vary with the square of the velocity; but at the same time the length of the path along which the steam must flow is being reduced by increasing the pressure drop, so that the question, when the friction is the smallest, is not an easy one to solve.

The various considerations suggested above explain why there is such a great difference in the number of wheels in the principal makes of turbine wheels. Professor Bull believes, however, that the difference will grow less and less as the years pass by and experience accumulates. It must be remembered that the successful steam turbine is but very young in years, and that it will take a number of years before experimenters and designers will agree as to the propositions, and even to the main ideas, in the design which will give the best results. There is room for a very large amount of experimental work of a very intricate nature and on a large scale to determine various problems in the working of the steam turbine, and until that has been done ideas will be more or less confused as to the actual behavior of the steam in the turbine, and also as to the friction of the steam along the vanes and of the friction of the wheels running in steam of various pressures.

Notwithstanding this lack of positive knowledge, it is not to be doubted, Professor Bull says, that the steam turbine is still very far from having reached its greatest state of perfection, and that it has a brilliant future before it. Not all of the designs have been successful by any means; in fact, quite a

few have been failures. It lies very near to ask the question: How was it possible for the steam turbine, notwithstanding the failure of many designs and of the apparent newness of the idea, to obtain such a marked success? The reasons are of various kinds. First of all, the idea was really an old one, and more work had been done than people in general were aware of. Second, the reciprocating steam engine had apparently reached the highest perfection which it was possible to impart to it, so that inventors had to turn their efforts into a new channel if they were to expect noteworthy results. It is true enough that the efficiency of the reciprocating steam engine has been increased considerably during the last fifteen or twenty years by means of the use of superheated steam, but this gain in economy is shared by the steam turbine as well, so that their relative position is not changed because of the use of superheated steam. A third reason must be sought in the fact that because of its purely rotative motion it avoids those irregularities of motion which are inherent in the reciprocating steam engine, even if it be provided with the heaviest fly wheel. A fourth reason for the rapid introduction of the steam turbine is, no doubt, to be found in the fact that it occupies much less space than its rival, a very important consideration in large cities and on shipboard. A further reason may be found in the fact that it can be regulated much more closely than the steam engine, this because of the very short time—only a very small part of a second—that the steam occupies in passing through the steam turbine, the corresponding time for the reciprocating engine being many times greater.

It will be noticed that an improvement in economy has not been given as one of the reasons for the rapid introduction of the steam turbine, and Prof. Bull is of the opinion that no such improvement has yet been shown. According to the best knowledge possessed, the present status seems to be that for the most favorable load the reciprocating engine still seems to have a slight lead, whereas for a variable load, as is most frequent in electrical lighting and traction plants, the opposite is probably the case. Prof. Bull holds that it is, however,

almost impossible to express a definite opinion in this respect, as a large part of the material upon which such an opinion must be based is furnished by the manufacturers of the turbines themselves, and which consequently cannot be taken as the very best kind of evidence. It is true that lately this kind of material has been furnished more and more by impartial observers, so that a comparison with the reciprocating steam engine, for which such a large amount of incontrovertible material is available, has become very much easier.

The reasons for the gain in economy, due to the use of superheated steam in reciprocating engines and turbines, are very far from being the same, and an analysis of these will help to furnish an estimate of what can be expected ultimately in this direction from the steam turbine. In the case of the reciprocating steam engine, the gain is due to the reduction of the cylinder condensation, which is due to the difference of the temperature of the saturated steam and of that of the cylinder walls and head, which condensation is greatly intensified by the moisture in the steam adhering to the metallic surfaces. Consequently the gain is not alone due to the fact that the superheated steam is capable of giving up a certain amount of heat without being condensed, but also—and in most cases to a greater extent—because very much less heat is transmitted from the superheated steam to the cylinder than if there has been saturated steam for the same difference of temperature.

In the case of the steam turbine, this difference of temperature is necessarily very much smaller, as no exhaust steam touches the parts which come in contact with the steam on its way through the turbine. On the contrary, if the barrel of the steam turbine is well protected, the temperature of the inside of this barrel can be but very little different from that of the steam at that particular place, hence, the condensation due to this cause is certainly small compared with that in the reciprocating engine. Because of the absence of moisture on the cooler surfaces touched by the steam this condensation will be reduced in the same ratio as for the reciprocating engine; but the difference of temperature being so small, this

gain, because of the absence of moisture, can only be insignificant. However, even in the ideal case of an adiabatic expansion, condensation is inevitable if saturated steam be used, and such condensation will be prevented if the steam be sufficiently superheated so that its temperature may be lowered without its becoming saturated. In this respect the steam turbine and the reciprocating steam engine are on the same level as to the possible gain due to the use of superheated steam. So far, the analysis has unmistakably shown that the last-named motor has the advantage; but there is an additional reason for the gain in economy by the steam turbine, and that is in the reduced friction which the superheated steam encounters on its way through the turbine as compared with the saturated steam. This friction is entirely negligible in the case of the reciprocating steam engine; but because of the very high velocities utilized in the steam turbine it is a very important factor in its economy. This friction is not alone due to the motion of the steam through the various wheels of the turbine, but also to the rapid revolution of the wheels in this steam. How much the friction amounts to it is at present impossible to say, although some experimental work has been done in this direction, notably by Dr. A. Stodola, of Zurich, Switzerland; but it amounts to a good deal, and there is no question but that it is very much smaller for superheated steam than for saturated.

The use of superheated steam for the reciprocating engine has been accomplished by all kinds of difficulties because of the troubles of lubrication and the expansion of the cylinder due to the high temperature. Although these difficulties have been overcome to a very large extent, yet ordinary practice has limited the superheat to within reasonable limits because of these reasons. In the case of the steam turbine these difficulties do not exist, so far as lubrication is concerned, and but to a very much smaller extent with reference to the expansion of the metal. It is, therefore, possible to use a very much higher degree of superheat for the steam turbine than for the reciprocating steam engine without any bad effects, and this in itself is a distinct advantage.

Thermodynamics and experience both show that if condensation can be prevented, the efficiency is a function of the differences of temperature between that of the entering steam and that of the condenser. Theoretically, it should be proportional to this difference, but for actual engines it does not increase so rapidly as this would indicate, and there is probably a limit for the higher temperature beyond which the increase would be so small that it would not pay to go, even if circumstances are otherwise favorable. But this limit has certainly not been reached yet, Professor Bull says, and consequently there should be a decided gain for the economy of the steam turbine in the use of more highly superheated steam than has been used so far, or which can be used for the reciprocating engines. Tests of turbines using superheated steam show a gain in economy of about 1 per cent. for every 8 or 9 degrees of superheat.

A steam turbine can, of course, be run noncondensing, and its efficiency, if designed for this purpose, is probably as good at that of the noncondensing engine. But it is a fact established by a large number of tests that it is very much more important to have a good vacuum for a steam turbine than for a reciprocating engine, because the gain in economy is very much greater. It is, therefore, not unusual to hear of 28-inch vacuum for a steam turbine, and such high vacuum has only become possible by the introduction of improved condensers and air pumps, which owe their origin to the demands of the designers of the steam turbines. Why the gain in economy because of a high vacuum should be so much higher for the steam turbine than for the ordinary steam engine, is a question which has troubled a great many persons. The answer is, however, quite simple. Because of the increased condensation it has been found unprofitable to extend the expansion down to the back-pressure line in the steam engine, so that the steam, when exhausted, possesses a considerable amount of energy over and above that at the pressure of the condenser, which energy consequently is lost. In the turbine this is entirely different, as the condensation is

so small that it does not need to be taken into consideration, and the energy of the steam is changed into a kinetic form, and which, in amount, is proportional to the difference in pressure between that of admission and of condenser. This energy can be utilized, except for friction losses, down to a certain minimum, corresponding to the necessary velocity of the steam when it leaves the last wheel, and which is the same, whether the vacuum is high or low. Although, therefore, there is gain in economy for the steam engine by increasing the vacuum, yet it cannot be so great as for the steam turbine, because the losses of condensation increase so much more rapidly than in the case of the turbine. Professor Bull points out that too much money may be paid for condensing apparatus, water and running expenses in order to get an exceptionally good vacuum; the extra expense may more than outweigh the gain in steam consumption.

It is interesting to note in this connection the success achieved by Rateau in utilizing the exhaust steam from a steam-hoisting engine at a mine for driving a low-pressure steam turbine. As the hoisting engine was only running intermittently, he inserted what he calls a steam accumulator—a mass of iron which absorbs heat during the time when there is an excess of exhaust steam, and gives it off by evaporating water, when there is a deficiency, in this manner enabling the turbine to run continuously. An intermittingly running engine like the hoisting engine can hardly be run condensing, because it takes too long a time to produce the vacuum. But a steam turbine with the accumulator can utilize the exhaust steam with a comparatively high efficiency, as is fully shown by the tests of such turbines in operation. In one test with a pressure of 14.71 pounds absolute the consumption of steam per electrical horsepower per hour was only 39.97 pounds per hour, and for a pressure of 9.37 pounds absolute it was 42.78 pounds.—"Engineering Record."

SHIPS.

UNITED STATES.

The Navy Department has assigned names to new vessels as follows:

Fleet Collier No. 1, *Erie* is changed to *Vestal.*
Fleet Collier No. 2, *Ontario* is changed to *Prometheus.*
Battleship No. 26, *South Carolina.*
Battleship No. 27, *Michigan.*
Sea-going Tug No. 10, *Patapsco.*
Sea-going Tug No. 11, *Patuxent.*
Submarine Torpedo Boat No. 9, *Octopus.*
Submarine Torpedo Boat No. 10, *Viper.*
Submarine Torpedo Boat No. 11, *Cuttlefish.*
Submarine Torpedo Boat No. 12, *Tarantula.*

Vermont—*Launch of.*—The battleship *Vermont* was successfully launched at the works of the Fore River Shipbuilding Co., Quincy, Mass., on August 31, 1905. This vessel is a sister ship to the *Kansas*, for description of which see page 887, Vol. XVII.

Mississippi—*Launch of.*—This vessel was successfully launched at the works of the Wm. Cramp & Sons' Ship and Engine Building Co., Philadelphia, Pa., on September 30, 1905.

The *Mississippi* is one of two sister ships, the other being the *Idaho*, authorized by Act of Congress, approved March 3, 1903. The contract for the construction of the *Mississippi* was signed January 25, 1904, the price for hull and machinery being $2,999,500.

The ship has a water-line length of 375 feet, a beam of 77 feet, and a normal draught of 24 feet 8 inches, the length over all being 382 feet. The displacement on trial, with a load including 600 tons of coal, is to be 13,000 tons, at which the

engines, designed to develop 10,000 horsepower, are expected to give a speed of 17 knots. With the full bunker capacity of 1,750 tons of coal and full stores and ammunition on board, the displacement becomes 14,465 tons. The machinery consists of twin-screw, triple-expansion engines, with cylinders measuring respectively 25½, 42 and 69 inches in diameter and a common stroke of 48 inches. These engines are operated by steam furnished by eight Babcock & Wilcox boilers, with a total grate surface of 768 square feet and heating surface of 32,640 square feet, this giving a ratio of 42.5 to 1. The estimated steaming radius at ten knots is 5,750 nautical miles.

The main battery consists of four 12-inch, eight 8-inch and eight 7-inch breechloading rifles and two 18-inch submerged torpedo tubes. The secondary battery consists of twelve 3-inch rapid-fire guns, six 3-pounders, four 1-pounders, two 3-inch field pieces, two machine guns and six automatic guns, caliber .30. The 12-inch guns, arranged in pairs, are in two electrically controlled, balanced, elliptical turrets on the center line, one on the upper deck forward and the other on the main deck aft, with an arc of fire of about 270 degrees and 250 degrees, respectively.

The hull of the *Mississippi* is protected at the water line by a complete belt of armor, nine feet three inches wide and of a uniform thickness of 9 inches for about 244 feet amidships, forward and aft, of which the belt is reduced in width and the thickness is gradually decreased to 4 inches at the stem and stern. Triangular athwartship armor in wake of water-line belt, is 7 inches uniform thickness. There is a complete protective deck extending from stem to stern, the deck being flat amidship, but sloped at the sides throughout and at each end.

Rhode Island—*Official Trials.*—The official preliminary trials of the battleship *Rhode Island* were held over the measured mile off Rockland, Maine, November 1, 1905, and the four-hours' trial in free route November 11, 1905. A detailed description of this vessel and her trials will appear in the next issue of the JOURNAL.

Louisiana—*Dock Trial.*—A successful dock trial of the machinery of the battleship *Louisiana* was held at the works of the Newport News Shipbuilding and Drydock Company from August 8 to 12, 1905. For a general description of this vessel see Vol. XVI, page 996.

Connecticut—*Dock Trial.*—The main engines of the United States battleship *Connecticut*, which is building at the Navy Yard, New York, were given a dock trial August 21 to 26, 1905. The engines were run separately on alternate days, three days being devoted to the trial of each engine. The actual running time for each engine was about twenty hours. During the first day's trial of each engine the revolutions averaged from 35 to 40, only two boilers being used. On the second day's trial the revolutions were increased to an average of about 60, four boilers being used. On the third and last day's run eight boilers were used, the revolutions averaging between 75 and 80, the maximum being, when the engines did not race, 86 for the port engine and 84 for the starboard one.

Indicator cards were taken at various speeds from each engine. The starboard engine required more horsepower for the same number of revolutions than the port one. This was due to the starboard side of the ship being next to the dock. The port engine showed a tendency to race, and at one time the revolutions reached 104 per minute without any increase in the steam pressure. With the same mean pressure upon the pistons as the cards show for the higher revolutions at the dock, the revolutions should be, in free route, a little over 120 per minute.

Although this was the first time the engines had ever been turned under steam they ran easily and smoothly, apparently being in perfect adjustment. No water was used on any of the bearings, nor allowed to circulate through any of the hollow brasses or guides, with the exception of the thrust bearings, when, owing to the increased thrust due to the ship being fast to the dock, water was allowed to circulate through the cooling coil and to a limited extent through the hollow

horseshoes. The results of this trial show the excellent character of the work done by the employés at the New York Navy Yard.

The principal dimensions of the *Connecticut* are as follows: Length, 450 feet; beam, 76 feet 10 inches; draught, normal, 24 feet 6 inches; draught, full load, 26 feet 9¼ inches; displacement, normal, 16,000 tons; displacement, full load, 17,666 tons; speed at normal displacement, 18 knots. The engines are designed for a total of 16,500 horsepower at 120 revolutions, with 265 pounds steam pressure at the boilers. They are of the usual vertical, four-cylinder, triple-expansion type, the high and intermediate-pressure cylinders being in the center, and one low-pressure cylinder at each end. The cranks for the forward pair of cylinders and also for the after pair are opposite to each other, while the two pairs are set at right angles. The framing is all of forged steel, consisting of vertical columns well trussed by suitable stays. Piston valves are used on all the cylinders, and are worked by the usual double-bar Stephenson link motion. The crank shafts are hollow, forged in two sections, each section having two cranks opposite to each other. This is done to keep the cylinders as close to each other as possible, and thus reduce the vibration.

The principal dimensions of the engines are as follows:

Diameters of cylinders, inches	32½, 53, and two of 61
Stroke, inches	48
Diameter of piston rods, inches	7¼
Length of connecting rods between centers, inches	96
Diameter at neck, inches	6¾
crank end, inches	8
of crank pins, inches	17¼
Length of crank pins, inches	19
Diameter of crank-shaft bearings, inches	16¼
Length of crank-shaft bearings, inches	19¼
Cooling surface of each condenser, square feet	10,375

The boilers are of the Babcock & Wilcox type, having 52,752 square feet of heating surface and 1,097 square feet of grate surface, a ratio of 48.1 to 1.

RESULTS OF TRIAL.

	Port engine.	Starboard engine.
Revolutions per minute	86	83
I.H.P., high-pressure cylinder, pounds	2,260.78	1,942.35
intermediate cylinder, pounds	2,232.11	2,221.42
forward low pressure, pounds	1,035.01	1,124.09
after low pressure, pounds	1,134.49	1,094.89
collective, pounds	6,662.39	6,382.75
Steam pressure, per gauge, pounds	265	255
First receiver pressure, absolute, pounds	98	98
Second receiver pressure, absolute, pounds	35	32
Vacuum, in inches	25.5	25.5
Total indicated horsepower	13,045.14	

—Extract from article by H. P. NORTON, in "Marine Engineering."

ENGLAND.

Natal—*Launch of.*—The armored cruiser *Natal*, launched September 30 by the Duchess of Devonshire from the Naval Construction Works of Messrs. Vickers, Sons & Maxim, Limited, is interesting, not only because this warship illustrates a step in the progress of cruiser design which will ultimately involve their assimilation with battleships, but in her name the *Natal* signalizes the recognition by one more of the colonies of a financial responsibility in the maintenance of the first line of Imperial defence. It is only quite recently that the Parliament of Natal agreed to make a contribution, and for the current year, out of £431,400 paid towards naval expenditure by the colonies, Natal finds £35,000, a fairly large sum per unit of size and wealth; the Cape Colony contributes £50,000; Australasia, £200,000; New Zealand, £40,000; India, £103,400; and Newfoundland, £3,000. The Admiralty's acknowledgement of this generous act by the naming of such a useful cruiser the *Natal*, and the instruction that the launching ceremony should be, as far as convenient, a colonial function, should induce the other members of the Empire to similarly recognize their responsibility in the maintenance of our sea power.

The *Natal*, as we have said, marks the transition stage in the evolution of almost a new condition so far as our warships

are concerned, and an analysis of her gun power shows the progress made. She mounts, like her three consorts—the *Warrior*, *Achilles* and *Cochrane*, all designed by Sir Philip Watts, K.C.B., Director of Naval Construction—six 9.2-inch and four 7.5-inch guns, as compared with the six 9.2-inch and ten 6-inch weapons in the immediately preceding cruisers for our Navy. She belongs to the first cruiser class in the British Navy, in which the 6-inch gun has been entirely dispensed with, and although this has involved a reduction in the number of guns, the gain is enormous in the penetrating power of each shot fired. The 6-inch gun at its best has a muzzle energy of 6,500 foot-tons; in the most of our ships it does not exceed 5,000 foot-tons. This makes it quite ineffective at the ranges naval tacticians now consider most suitable even in what have hitherto been regarded as essentially cruiser tactics. The 7.5-inch gun, on the other hand, if constructed of 50-calibers in length, as is the case with those in the *Natal*, develops a muzzle energy of over 11,500 foot-tons; and as the possible rapidity of fire per minute is nine shots for the 6-inch gun to six in the 7.5-inch weapon, the advantage, from this point of view, of the lighter weapon does not compensate for its ineffectiveness in power. The 9.2-inch gun, however, is now accepted even in preference to the 7.5-inch piece, and here the 50-caliber weapon may develop a muzzle energy of 24,000 foot-tons. Assuming, therefore, favorable conditions so far as rapidity of fire is concerned, we find that the *Natal* will develop practically 835,000 foot-tons of muzzle energy for each minute's fire, as compared with 829,000 foot-tons in the preceding ships; while but a few years ago our best cruisers, those of the County class, fitted only with 6-inch guns, developed a total of only 542,000 foot-tons. The important point, however, is that the *Natal* may fire per minute 48 shots, which can be effective at 3 miles range, as compared with 24 in the case of the immediately preceding ships—those of the *Duke of Edinburgh* class, eight in the case of the *Drake*, ten in the *Devonshire*, and none in the County class. The following table is instructive:

FIRE OF PRIMARY ARMAMENT PER MINUTE.

1905. *Natal* (13,550 tons).	24	at 380 pounds =	9,120	pounds and	580,560 foot-tons.
	24	200	4,800		254,400
	48		13,920		834,960
1904. *Duke of Edinburgh* (13,500 tons).	24	380	9,120		441,600
	80	100	8,000		387,200
	104		17,120		828,800
1901. *Drake* (14,100 tons).	8	380	3,040		147,200
	128	100	12,800		619,520
	136		15,840		766,720
1904. *Devonshire* (10,400 tons).	10	200	2,000		101,200
	80	100	8,000		387,200
	90		10,000		488,400
1900. *Kent* (9,800 tons).	112	100	11,200		542,080

Another important improvement in connection with the armament which Sir Philip Watts has introduced in the ships for our Navy is the greater height of the gun platform above the sea level. In the *Natal* and her sister ships all the guns are on the upper-deck level. In gun houses right forward and aft there is located a 9.2-inch weapon, encased in a barbette of 6-inch armor, decreasing to 3-inch at the base. On the four quarters of the citadel there are similar 9.2-inch guns close to the broadside, and, as the forecastle structure is cut away, these guns fire in line with the keel, and have a very considerable arc of fire—180 degrees. Four 7.5-inch guns are placed within casemates on the upper deck, one on each side, near to the center of the length of the ship, the protecting armor here also being 6 inches in thickness. All the guns are loaded and the mountings rotated by hydraulic power. The equipment for dealing with torpedo or submarine attack includes twenty-three quick-firing guns, two 12-pounders, and two Maxim guns. The hoists for supplying all ammunition for these guns are operated by electric motors. In addition, the vessel has two under-water 18-inch broadside torpedo tubes, and one submerged 18-inch torpedo tube firing astern.

The armor is very effectively distributed. There is, in the first place, a complete broadside belt from bow to stern, ex-

tending from about 5 feet under the water line to the upper deck in the center part of the ship. The thickness of the armor over the machinery spaces, &c., extending over 60 per cent. of the length of the ship, is 6 inches; forward of this it narrows, and is thinned down by two stages, first to 4 inches, and ultimately at the extreme end of the ship to 3 inches in thickness. The protective deck proper extends from the stem to the stern, and ranges in weight from 80 pounds to 30 pounds per square foot. This deck is worked at the lower edge of the armor to assist in the protection of the vital parts of the vessel. Another deck of 40 pounds per square foot forms a crown for the side armor and the armor bulkheads forward and aft, these latter being 6 inches in thickness. The conning tower is 10 inches in thickness, and is provided with an armor tube for the lines of communication to below the protective deck. For communication electric telephones are fitted, and the lighting is also by means of electric power, the double-wire system of distribution being adopted. There are six powerful searchlights. Wireless telegraph equipment will be included.

The dimensions of the ship are as follows: Length between perpendiculars, 480 feet; breadth, extreme, 73 feet 6 inches; displacement at 27 feet draught, 13,550 tons. The normal coal supply carried is to be 1,000 tons, but bunker capacity has been provided for a much increased supply.

The propelling machinery of these cruisers has, as far as possible, been standardized, the general system adopted being that which experience has proved to be most satisfactory for cruisers. The cylinders, all independent castings, are arranged on the Yarrow-Schlick-Tweedy system of balancing, being placed in couples close together, with the valves to the outside; they are supported on an A frame at the rear, with forged-steel columns in front. There are four cylinders arranged for triple compounding, the diameter of the high-pressure cylinders being 43½ inches; of the intermediate, 69 inches; and of the two low pressures, 77 inches. Piston valves are used for both high and intermediate cylinders, and slide valves for the two low-pressure cylinders. The stroke is 42 inches, and

the total power of 23,500 indicated horsepower is expected to be realized with the machinery running at 135 revolutions. The crank, thrust and propelling shafts are of forged steel, and hollow. The two propellers are of manganese-bronze, with adjustable blades, and work outwards when the vessel is going ahead. The main condensers, oval in section, are of gun metal, with brass tubes, there being two to each set of engines, making four in all, each with its own centrifugal pump of gun metal, with independent engine. These condensers deal also with the steam from the auxiliary machinery of the ship, an arrangement which greatly simplifies the planning of the engine room. The air pumps are entirely separate, and are of Weir's well-known type.

The boiler installation was decided upon before the Admiralty had come to the conclusion that the combination of cylindrical and water-tube boilers had disadvantages which more than made up for any possible economy which, it was assumed, might be derived by the installation of the tank boiler for cruising purposes The *Natal* is fitted with twenty Yarrow boilers and six cylindrical boilers, two-thirds of the power being represented by the former. The proportion of power to weight of machinery is about 10½ horsepower per ton, which is lower than is attainable now with a complete installation of water-tube boilers. The speed anticipated by the new ship is 22.33 knots, and the complement of officers and men is about 700. The *Natal*, it may be added, is in a very advanced stage of construction, the launching weight being 7,400 tons, of which 300 tons is due to the cradle, the practice at the Vickers Works being to defer the launch as long as possible, as there is no difficulty in launching a heavy hull which has been well advanced and clad in armor before being floated. This latter is an economical procedure.—"Engineering," London.

Argyll—*Trials.*—The armored cruiser *Argyll*, belonging to the *Devonshire* class, built and engined by Scotts' Shipbuilding and Engineering Company, of Greenock, has completed her official steam trials with satisfactory results. This cruiser, which has already been described (see JOURNAL A. S.

N. E., Vol. XVII, p. 430), has a length of 450 feet, a breadth of 68 feet 6 inches, a depth, molded, of 38 feet 6 inches, and at a mean draught of 24 feet 9 inches displaces 10,700 tons. It will be noted that at the trial the mean draught in each case exceeded that designed, but this was due to extra load of water ballast, and is only mentioned here to show that the trial steam performances will be maintained when the vessel has all her war stores on board. On the trial she had all her guns, and was practically ready for service; indeed, the gun trials formed part of the program of her steam tests. The vessel proceeded on her preliminary trial to the South coast, where the ordnance tests were carried out in the English Channel; subsequently she returned to the Clyde to complete the official performances.

The machinery is of the four-cylinder, triple-expansion type, the high-pressure cylinders being 41 inches in diameter, the intermediate cylinders 65½ inches, and the two low-pressure cylinders on each engine 73½ inches, having a stroke of 42 inches. There are sixteen Babcock & Wilcox, and six cylindrical boilers, arranged in four compartments.

STEAM TRIALS OF H. M. S. *ARGYLL*.

Date of trial			Sept. 15, 16, 1905.		Sept. 19, 20, '05.		Sept. 26, 1905.
Nature of trial			30 *a*		30 *a*		8 *b*
Draught of water (mean), feet and inches.			Fwd. 24–5	Aft. 25–7	Fwd. 24–6	Aft. 25–6	Fwd. 24–6	Aft. 25–6
Speed, knots			13.9 *c*		20.8 *d*		22.28 *d*
Steam pressure in boilers, lbs. per sq. in.			180		202		194
Air pressure in stokeholds, in. of water			Nil		0.6 *e* 0.4 *f*		0.97 *g* 1.05 *h*
Vacuum in condensers, inches			Stbd. 25.25	Port. 25.5	Stbd. 24.7	Port. 24.7	Stbd. 24.5	Port. 25.1
Revolutions per minute			85.8	87	125.7	126.6	138.9	138.9
Mean press. in receivers.	High, lbs.		110.5	107.8	171	170	172	173
	Inter., lbs.		17.2	16.1	57	53	68.3	67.8
	Low, lbs.		9	11.8	9.2	7.2	15.6	14.3
Mean pressure in cyls.	High, lbs.		35.9	36.9	60.9	62.8	63.3	68.6
	Inter., lbs.		12.2	13.5	27 8	28.6	36.76	37.85
	Low, fore, lbs.		4.1	4.28	11.7	11.7	16.38	15.75
	Low, aft, lbs.		4.39	4.31	12.9	12.3	18.44	17.6
Mean I.H.P.	High		884	920	2,197	2,282	2,522	2,736
	Intermediate		752	841	2,503	2,596	3,652	3,760
	Low, fore		317	334	1,322	1,336	2,047	1,969
	Low, aft		339	339	1,467	1,405	2,305	2,199
Total I.H.P.			2,292	2,434	7,489	7,619	10,526	10,664
Grand total			4,726		15,108		21,190	
Consumption of water (all purposes) per I.H.P. per hour, pounds			20.21		17.28		18.91	
Consumption of coal (all purposes) per I.H.P. per hour, pounds			1.94		1.82		2.2	

a, Hours' coal consumption. *b*, Hours' full power. *c*, (By bearings.)
d, (Mean of four runs.) *e*, Water-tube. *f*, Cylindrical.
g, Cylindrical boilers. *h*, Water-tube boilers.

On the first trial of thirty hours' duration the vessel was required to develop one-fifth of the total horsepower, and it will be noted from the accompanying table that the mean result of the hourly readings was 4,726 indicated horsepower, the consumption of water per indicated horsepower for all purposes being 20.21 pounds. It should be noted in this connection, however, that the proportion of steam used by the auxiliary machinery to the power of the main engines is very considerable. The coal consumption for all purposes was 1.94 pounds. On the second trial, also of thirty hours' duration, the engines developed 15,108 horsepower, and the consumption of water for all purposes was 17.28 pounds, while the fuel consumption was 1.82 pounds. Four runs were made over the measured mile to ascertain the speed, and it was found that this rate worked out to 20.8 knots. On the full-power trial, which took place on Tuesday of this week, the mean draught was 25 feet, and the power developed was 21,139, or 190 indicated horsepower in excess of that guaranteed by the builders. In this connection it should be remembered that the builders of British warships designed at the Admiralty, as in the case of the *Argyll*, are not responsible for speed.

The *Argyll*, on four runs on the measured mile, made a mean speed of 22.38 knots, while the speed anticipated in the design was 22¼ knots; the state of the hull, due to the fact that the vessel had not been in dock for three months, and the problem of propeller design probably accounts for the fact that this vessel has not steamed at quite the same high rate as the other cruisers of the class. Indeed, as we have already pointed out, the Admiralty are accumulating data as to the comparative performances of Admiralty propellers in the vessels of this class, and it may be that the design of the screws of the *Argyll* have not yielded as satisfactory results as those on other vessels of exactly the same design and power. It will be noted that the air pressure in the stokehold on the full-power trial was about 1 inch, and that the consumption of water was 18.91 pounds. The total loss of water during the

eight hours' trial was 16.8 tons, which is equal to 2.37 tons per 1,000 I.H.P per day of twenty-four hours.

The full results are tabulated in the accompanying table, and an examination of these details will show that they are, as far as the machinery is concerned, most satisfactory. Circling, stopping, starting and gun trials have also been carried out, and the vessel will be at once completed, at the works of her builders, for commission.—"Engineering," London.

FRANCE.

Jules-Michelet—*Launch of.*—The first-class armored cruiser *Jules-Michelet* was launched on the 31st August from the dockyard at Lorient. Her dimensions are as follows: Length, 489 feet; beam 70 feet 2 inches; displacement, 12,750 tons, on a mean draught of 26 feet 3 inches. Protection is afforded by a complete armor belt of hard steel, with a thickness of 5.9 inches in the center, and 3.94 inches at the extremities, extending 7 feet 6 inches below the water-line, over a plate thickness of 0.79 inch. Above the belt the armor will have a thickness of 4.9 inches, to a height of 7.54 feet above the water line. Forward, this armor will be carried as high as the casemates of the 6.48-inch guns. The two armor decks will start respectively from the top and bottom of the armor belt, and will form with it a caisson divided into a number of watertight compartments. The armament will consist of two 9.4-inch guns in the fore turret, and two 7.6-inch in the after, the turrets being protected by 8-inch armor, with 5-inch hoists; twelve 6.48-inch guns in casemates; and a number of 2.56-inch and 1.85-inch guns; also five torpedo tubes, two of them submerged. Her three engines of triple expansion will develop a total of 29,000 H.P. She will have twenty-eight Du Temple-Guyot small-tube boilers, and four funels, 69 feet in height. Her speed will be about 22.5 knots. When launched, she was already fitted with her propeller shafts and her false keels, and her upper decks were finished. Her launching weight was 5,000 tons, representing 60 per cent. of her construction. She

was laid down 1st June, 1904, and should be ready for her trials in 1907. Her cost is estimated to be 30,842,731 francs.

In the case of her sister ship, the *Jules Ferry*, some modifications were introduced in the designs of the boilers, by which the grate surface was augmented and the coal consumption per square foot of grate surface per hour was reduced from 44.64 pounds to 38.3 pounds; in the boilers of the *Jules-Michelet* the proportion of grate surface to heating surface has been still further reduced, and, at the same time, the number of tubes in a boiler has been decreased by one-twentieth, with the result, it is hoped, of reducing the coal consumption to 33.38 pounds per square foot of grate surface per hour.—Exchange.

Submarine Y—*Launch of.*—The new submarine *Y* was launched on the 23d July at Toulon; the order for her construction was given in August, 1901, but she was not actually begun until 22d May, 1902. The *Y* was designed by M. Bertin; she has a displacement of 213 tons, is 140 feet 6 inches long, with a beam of 10 feet 6 inches. She is fitted with explosion motors of 250 H.P., which will be used both for moving on the surface as well as when submerged, and is to have a speed of 11 knots. She is fitted with four torpedo discharges.

GERMANY.

Elsass—*Trials.*—The first-class battleship *Elsass* maintained a mean speed of 18 knots during her six hours' forced-draft trial, the engines making 112.6 revolutions and developing 16,685 I.H.P.

During the measured-mile trials 18.7 knots speed was obtained with 16,812 I.H.P. and 114.9 revolutions per minute.

The coal consumption for a twenty-four hours' trial was found to be 1.73 pounds per I.H.P. per hour, the vessel making 16.5 knots and developing 11,559 I.H.P.

Preussen—*Trials.*—The first-class battleship *Preussen* has recently completed her trials. With the engines developing 18,374 I.H.P., and making 116.7 revolutions, a mean speed of 18.6 knots was attained on the measured mile. With the

engines developing 9,717 I.H.P., and making 97.7 revolutions, a mean speed of 16.4 knots was maintained. On the twenty-four hours' coal-consumption trial, with the lateral engines developing 6,800 I.H.P., the coal consumption was 1.58 pounds per indicated horsepower per hour.

München—*Trials.*—The third-class cruiser *München* has completed her trials successfully. At her twenty-four hours' full-speed run with the engines making 121.65 revolutions and developing 7,164 I.H.P., a mean speed of 20.4 knots was maintained with a coal consumption of 1.92 pounds per I.H.P. per hour; at a coal-consumption twenty-four hours' trial with the engines making 122 revolutions and developing 7,238 I.H.P., the mean speed was 20.4 knots, and coal consumption 1.92 pounds per I.H.P. per hour. At the full-speed trial on the measured mile, with the engines making 143 revolutions and developing 12,388 I.H.P., the mean speed was 23.45 knots.

Lübeck—*Trials.*—The new third-class cruiser *Lübeck* has been undergoing her trials. She is fitted with turbine engines, but the success attained did not equal expectations, as the speed was not greater than that of her sister ships, the *Hamburg* and *Bremen*, which are fitted with the ordinary reciprocating engines, while the consumption of coal, up to a speed of 16 knots, was greater. The trials were carried out with two small screws, fitted to each of her four propeller shafts, and she is now to go into dock to be fitted with larger screws. In changing from going ahead to going astern, the *Lübeck* was slower than her sister ships with pistons. The engines of the *Lübeck* required considerably less oil, which compensates somewhat for her greater coal consumption at moderate and slow rates of speed. During a twenty-four hours' run, with the engines developing 7,600 I.H.P. and making 670 revolutions, a speed of 20.5 knots was maintained, with a coal consumption of 0.9 kg. (1.98 pounds) per I.H.P. per hour, with the engines developing 9,500 I.H.P., a speed of 21.5 knots was maintained, with a coal consumption of 1,075 gr. (2.149 pounds) per I.H.P. The weight of the turbine engines is 77.5 tons less than the reciprocating engines of her sister ships, and

the ship seems to be steadier in a seaway. It is interesting to note that in the British third-class cruiser *Amethyst*, which is fitted with turbines, the economy of fuel was progressive from 14 knots up to full power, when she completely distanced her sister ships, the *Diamond* and *Topaze*, both in speed, power and economy of fuel.—Exchange.

Berlin—*Trial of.*—The third-class cruiser *Berlin*, during a twenty-four hours' trial, maintained a mean speed of 21 knots. The engines developed 7,121 I.H.P. and averaged 115 revolutions per minute.

A mean speed of 22.6 knots was obtained during the six-hours' forced-draft trial, the corresponding I.H.P. being 10,857, with the engines averaging 127.6 revolutions.

A speed of 23.26 knots was reached on four runs over the measured mile.

Hannover.—The following are the particulars of the German battleship *Hannover*, recently launched at Wilhelmshaven: Length over all, 430.289 feet; between perpendiculars, 398.628 feet; beam, 72.83 feet; draught, 25.26 feet; engines, triple-expansion, of 16,000 horsepower; boilers, Schultz-Thornycroft, to the number of twelve; speed, 18 knots; armament, four 11-inch guns in two turrets with 11-inch protection; fourteen 6.7-inch guns, ten in the central casemate and four in four separate casemates; twenty-two 3.46-inch and fourteen 1.45-inch guns, four mitrailleuses, and six torpedo tubes, submerged; thickness of armor belt amidships, 9.45 inches decreasing to 5.9 inches, and then to 3.93 inches fore and aft; thickness of armor above the belt, 8 inches; coal, ordinary load, 700 tons; special load, 1,800 tons.—"Engineering," London.

Danzig—*Launch of.*—The third-class cruiser *Ersatz Alexandrine* was launched on September 23d from the Imperial Dockyard at Danzig, and received the name of *Danzig*. Her dimensions are as follows: Length, 340 feet 6 inches; beam, 43 feet 4 inches; displacement, 3,250 tons, on a draught of 16 feet 5 inches. There will be a 2-inch armored deck, and the conning tower will also be protected by 3-inch armor. The

armament will consist of ten 4.1-inch Q.F. guns, ten 1.45-inch Q.F. guns, with four machine guns and two submerged torpedo tubes forward for 18-inch torpedoes. The engines are to develop 10,000 I.H.P., giving a speed of 22 knots, while the coal capacity is 800 tons, giving a radius of action of 5,000 miles at economical speed.

ITALY.

Napoli—*Launch of.*—The new Italian battleship *Napoli* was launched on September 10th, at Castellamare. The *Napoli* will form a division of four homogeneous ships with the *Vittorio Emanuele*, *Reginia Elena* and *Roma*, but this last will not be launched until this time next year. The *Napoli* was laid down on October 21, 1903. Her length over all is 474 feet; between perpendiculars, 435 feet; beam, 82 feet; draught, 25.8 feet; displacement, 12,624 tons; number of engines, two; number of cylinders in each engine, four; number of boilers (Babcock & Wilcox), twenty-two; engines, 15,200 horsepower under natural draft, and 19,000 horsepower under forced draft; speed, probably 22.5 knots; ordinary load of coal, 1,000 tons; special load, 2,000 tons; radius of action at 12 knots with 2,000 tons of coal, 10,000 miles; armament, two 12-inch, twelve 8-inch, sixteen 3-inch, two 2.95-inch and ten 1.85-inch guns, two mitrailleuses, and four torpedo tubes. Her armor belt will run her whole length and be 9.18 feet broad, with a maximum thickness of 9.84 inches. Protection for 12-inch and 8-inch guns, 7.87 inches; for small guns, 3.15 inches; for conning tower, 9.84 inches. Her complement will be 36 officers and 679 men, or, when serving as flagship, 44 officers and 760 men.—"Engineering," London.

Submarines.—The Italian submarine *Glauco* has just been launched at Venice. This submarine, which is of a new type, has been built in succession to the *Delfino* and the *Triton*, which have not developed altogether satisfactory results. The *Glauco* will have a surface speed of 14 knots, and her radius of action will be 2,000 miles. Four other submarines are in

course of construction for the Italian Navy; they will be named the *Squalo*, the *Narvalo*, the *Ottaria*, and the *Tricheco*. —"Engineering," September 8, 1905.

JAPAN.

Mikasa—*Accident to.*—On September 11th, a fire broke out on the battleship *Mikasa;* it spread rapidly, reaching the after magazine, which exploded, causing the vessel to sink in shallow water. The loss of life was heavy, amounting to nearly 600, amongst whom were many from other ships who had gone to the rescue.

The cause of the fire is not known nor has the extent of the damage been determined, though it is known that a hole was blown in the ship's port side.

The *Mikasa* is a first-class battleship, having a displacement of 15,200 tons, and speed of 18 knots.

She was the flagship of Admiral Togo during the whole of the Japanese-Russian war.

SPAIN.

Cardenal Cisneros—*Loss of.*—The cruiser *Cardenal Cisneros* sunk on October 30 at 6 o'clock off. Muros, on the Galician coast. She was steaming to Ferrol and struck on a rock, owing to the fog. Her commander and all the officers and crew of 540 men were rescued, but the ship could not be refloated. The *Cardenal Cisneros* was an armored cruiser of 6,889 tons and 15,000 horsepower. She was built in 1902 at Ferrol, Spain, was about 364 feet long, had 60¼ feet beam, drew 25 feet of water and cost $3,000,000. The cruiser had 8 inches of armor and her armament consisted of two 9.4-inch guns, eight 5.5-inch guns and a number of smaller guns.

SWEDEN.

Magne—*Trials.*—The torpedo-boat destroyer *Magne*, built by Messrs. John I. Thornycroft & Co., for the Swedish Government, has recently completed her trials and has been sent to Sweden.

The vessel, which has a length of 219 feet 9 inches, with a beam of 20 feet 9 inches, was designed to attain a speed of 30.5 knots on a three-hours' trial when carrying a load of 50 tons, and for this purpose was fitted with two sets of four-cylinder, triple-compound, condensing engines, designed to develop 7,500 indicated horsepower, steam being supplied by four boilers of the Thornycroft-Schulz type. Generally, the vessel and the machinery correspond to the Japanese destroyers *Shirakumo* and *Asashio*, which were built by Messrs. Thornycroft, and proved, during the recent war, most satisfactory vessels, even in a very heavy sea. The full-power trial of the *Magne* was carried out on August 31st; under the required conditions of load, the vessel attained a speed during the three-hours' test of 30.705 knots, with the engines running at 378 revolutions per minute, and indicating 7,700 indicated horsepower.

MISCELLANEOUS.

Iris.—The following extracts are from a description of the motor launch *Iris*, in "Engineering," London, the various illustrations having been omitted. The *Iris* was designed for a sturdy type of sea-going launch, and built in accordance with the rules of the Marine Motor Association. The hull is built on the plain carvel system, and copper fastenings are used throughout. The following are the principal dimensions of the boat:

Length over all, 30 feet; length, water line, 29 feet; beam, extreme, 6 feet; beam, water line, 5 feet 8 inches; depth amidships, 3 feet 8 inches; freeboard, forward, 3 feet; freeboard, amidships, 2 feet 1 inch; freeboard, aft, 2 feet 2 inches; draught, extreme (at skey), 2 feet; displacement with crew, 2 tons; area, midship section, 4 square feet; block coefficient, 0.451; prismatic coefficient, 0.620; brake horsepower, 24; revolutions per minute, 800.

Speed being a secondary consideration, a very stanch craft was produced. Sea-going qualities and handiness being necessary qualifications, the midship section was kept very large; this, with a wide afterbody, has made her a very handy and

stable sea boat. The accommodation is large, seating ten persons aft quite comfortably, and providing ample storage space in the fore compartment, which is 6 feet long. The hull is of pine planking $\frac{5}{8}$ inch thick, with teak top strake. The stern dead woods and knees are of English oak cut from grown crooks. Running the whole length of the boat is an American elm keel inside and an Oregon pine keel outside, which forms the principal girder of the boat's frame, and is carried from the stem to within 5 feet of the stern. This keel is 3 inches thick throughout, has a depth of 15 inches at after end, tapering to the scarph of the stem, and secured to the inner hog keel by metal bolts through all. This makes the boat exceedingly rigid and enables her to take the hard ground without damaging the propeller or rudder; at the same time it adds considerably to her sea-going qualities.

The gunwales are in one length, each $2\frac{1}{2}$ inches by 2 inches, of Oregon pine, and are covered with a teak capping, running the whole length of the boat, and below this is a heavy Oregon pine rubbing strake. The motor and reversing gear being mounted on such a rigid channel frame, shallow motor bearers can be fitted, and these are 3 inches wide by 17 feet long, of selected red pine. They are bedded over six 3-inch by 2-inch oak floors, which fit the skin of the boat and extend well up the turn of the bilge. The stern tube is of gun metal, 5 feet long, and has a stuffing gland inside.

Fuel is supplied to the carburetter by gravity from a 10-gallon tank in the fore compartment under the turtle deck. There is a second fuel tank of 21 gallons capacity athwartships under the main thwart, and the fuel contained in this can be pumped into the forward, or running, tank by a semi-rotary hand pump placed conveniently close to the driver. The motor is encased in a neat mahogany housing with hinged tops, and copper trays are fitted under all the mechanism. A canvas spray hood is also fitted for heavy weather, and extends from the coaming at the aft end of the turtle deck to the steering column.

The *Iris* was engined by Messrs. Legros & Knowles, Limited,

of Willesden Junction, London, N. W. The engine is a four-cylinder petrol motor of 4¾-inches bore by 4¾ inches stroke, designed to develop 24 horsepower at 800 revolutions, at which speed the propeller normally turns. The inlet valves are mechanically operated, and are situated in the cylinder head; they are worked by rocking levers from the same cam shaft that operates the exhaust valves. On the other side of the engine there is a second cam shaft, which operates the mechanical make-and-break of the low-tension magneto ignition. High-tension ignition is also provided as a standby; this is done to eliminate the chance of a motor stoppage due to ignition trouble in a seaway. A special feature of the engine is the very large bearing surface given to the main bearings and crank pins, the mean pressures on these being kept as low as standard marine practice, and consequently far lower than is usually the case in petrol engines. The engine is also extremely accessible. It is held to its bedplate by three lugs on each side in the form of hinge-joints, thus enabling it to be swung to one side or the other to get at the big ends or main bearings. Two large and instantly-detachable inspection doors are fitted to the base chamber, through which the hand may be put on to any big end or main bearing. The two cam shafts are held up to their bearings by caps, and these bearings can be removed from the rest of the engine without disturbing the cylinders or base chamber.

The crank shaft is also held up into the engine bed, so that when the engine is swung on one side, and the oil-bath casing removed, the big ends and main bearings are immediately accessible for adjustment.

By means of the design of the carburetter, which gives a positive mixture for every position of throttle opening, the engine will run and pull steadily at all speeds from 140 revolutions upwards. The torque is practically constant throughout the range.

The circulation of water is maintained by a centrifugal pump, which is carried low down forward and driven direct from the crank shaft by a roller chain. The water is deliv-

ered first to a water-cooled exhaust pot carried close to the exhaust outlet, and from there proceeds to the engine jackets, and afterwards to the outlet overboard.

The valves are of very large diameter, of 25 per cent. nickel-steel. The crank shaft is a chrome-vanadium steel forging. From the engine the power is taken through a leather-faced cone clutch, which is engaged by an adjustable coil spring, and disengaged by a pedal. The clutch transmits the power to a small gear box, giving a forward and reverse gear by means of three miter bevels and a sliding sleeve with dogs, which enables the two shafts to be dogged together, or the power transmitted through the miter wheels. The power is transmitted through the bevels only when running astern. This gear box forms an oil-tight case, in which the bevels run, and which also lubricates the large phosphor-bronze bearings. It also has a platform cast on it, to which is fixed a cast-iron steering and control column. On this is mounted a vertical marine steering wheel, and in a convenient position for the steersman's right hand are two small handles, controlling, respectively, the throttle of the carburetter and the forward and reverse mechanism. The starting gear is also carried in this column. A removable handle fits into a socket in the column at a convenient height, and by means of a heavy chain and disengaging dogs the motor can be started. When once going the dogs disengage, and the chain does not run.

The after end of the engine housing forms an instrument board, on which is placed a pump lubricator, which is driven from a worm on the commutator spindle, and, consequently, delivers lubricant in quantities proportional to the speed of the engine. The two switches for magneto and high-tension ignition, the coil for the latter and the advance for both ignitions are also placed here.

Once the engine is started, the ignition may be advanced, and left without further adjustment during a run. Thus the whole control of the boat is easily managed by one man at the steering column, who can maneuver with great facility without moving from his place, and can re-start the motor, should he stop it. The engine is placed on its bearers with the

crank-center line horizontal, so that the oil in the bottom half of crank case may be equally distributed to the forward and after pairs of cylinders. This necessitates a Hooke's joint behind the gear box to connect with the tail shaft. The joint is made with the center line of both pairs of pins in the same plane; the pins are of nickel-steel, running in phosphor-bronze bushes. This joint runs quite cool and gives no trouble. The tail shaft is of delta metal, and is fitted with two ball-bearing thrust blocks and a gun-metal three-bladed propeller 17 inches in diameter. The whole installation of engine, gear box, steering column and control is lined up and fixed to a bedplate consisting of 4-inch channels, thus making a complete unit in itself, which can be fitted in a boat without the more difficult and often unsatisfactory operation of lining up in place.

The boat was completed in July, and ran in the Motor Yacht Club's reliability trials. These trials consisted of two days' running of ten hours each without stop, and an allowance of only ten minutes between the two runs to attend to the motor. The course was mainly in Southampton water, but there was an optional open-sea course in the Solent, round which this boat ran.

The first day the *Iris* ran absolutely without any stop and with great regularity for the ten hours in a smooth sea.

The second day, owing to the driver omitting to keep his front tank filled, the motor stopped, but it took only four minutes to find the cause, pump in petrol, and re-start the motor, though it was very rough at the time. On this day it was so rough that the trials were stopped after five hours' running, and the boat showed her great seaworthiness on the open course. She was the only one of her class (30-footer) which could be driven through the seas at her full speed without slowing down at all, and, thus driven, shipped no green water, but only spray. The reduction in speed due to the rough water was only 6.3 per cent. over the calm day, whereas reductions of speed up to 43 per cent. were shown by some of the other boats.

Since the trials the boat has been repeatedly navigated

single handed, and has proved herself exceedingly handy—so much so that, with a crew of only two, a tow-rope was passed to another boat which was towed across the Solent in rough water. Her speed is 10½ knots.

Fathomer.—The Coast Survey steamer *Fathomer* was recently placed in commission at Manila, and will be employed in the extensive surveying operations that are necessary to provide proper charts of the Philippine Islands.

The *Fathomer* was built by the Hongkong & Whampao Dock Co., at Hongkong, under a contract with the Philippine Government. The vessel was constructed under Lloyd's inspection as well as the inspection of the representative of the survey. Because of Lloyd's requirements nearly all the material had to be brought out from England. The contract was signed January 27, 1904; the trial trip was had on December 12, 1904, and the vessel was delivered in Manila Bay on January 19, 1905. The contract price of the vessel at Hongkong was $66,550, and, including delivery in Manila Bay, and extras but not equipment supplies, the total cost was about $70,000.

The *Fathomer* is a steel vessel of 368 gross tons; length over all, 160 feet; breadth, 25 feet; mean draught, 10 feet. The hull is divided into seven watertight compartments. In place of a double bottom, the lower decks, both forward and aft of the machinery space, are of steel and watertight, and there is special strengthening above Lloyd's requirements under the boiler and engine-room space. The mechanical equipment was made simple, because of the intention to employ mainly a native crew. A Scotch boiler and compound engine furnish the required speed of 10 knots. There is provided a dynamo, evaporator, small refrigerating machine and steam windlass.

Special attention was given to the boat equipment, an important feature in the success of surveying work. The *Fathomer* carries a 28-foot steam launch, a 24-foot whaleboat, a 20-foot cutter, a 16-foot dinghy and a 12-foot punt or flatboat. Wellin davits are provided for the launch, and the falls lead to the steam windlass.

MERCHANT SHIPS.

Maheno.—The turbine-driven steamer *Maheno* has just been completed by Messrs. William Denny and Brothers, Dumbarton, for the Union Steamship Company of New Zealand, and is the second turbine-driven steamer possessed by them. Her dimensions are: Length, 400 feet; breadth, 50 feet; depth, 33 feet 6 inches molded to upper deck; and she has almost a complete shade deck, with a boat deck over a large portion thereof. She is elaborately fitted out for passengers, of whom she will carry 223 first class, 116 second and 60 third class. The trial had to be carried out with not less than 300 tons of dead weight on board, but there is capacity for more cargo. At her first official trial, at full power, with all boilers in use, the *Maheno* easily attained a mean speed of 17.5 knots, and at the second trial, which had to be done with a third of the boilers shut down, the speed maintained for six hours was 16.4 knots—considerably over the guarantee. The vessel is propelled by a set of Parsons patent turbines, constructed by Messrs. Denny and Co., Dumbarton. There are three turbines, one high and two low pressure, working three shafts, with three propellers in all. The go-astern turbines are contained within the low pressure, and work on the wing shafts. The reversing gear is particularly well arranged, the whole of the engines being easily controlled by one engineer. Steam is supplied by four cylindrical tubular boilers, two double-ended and two single, at a working pressure of 150 pounds per square inch, and fitted with Howden's system of forced draft. The auxiliary machinery is very complete, and consists of air and circulating pumps to each main condenser, each set consisting of two Edwards air pumps and one centrifugal pump, driven by simple two-cylinder engines; two of Weir's feed pumps, auxiliary condenser with necessary pumps, two large duplex pumps for bilge and ballast purposes, a Caird and Raynor distiller, and a vertical duplex wash deck and

fire engine, suitable for working. See's ash ejector, sanitary pump, steward's pump, water-service pump, oil pumps, &c.—"Engineering," London.

Princesse Elisabeth.—A new turbine steamer has been added to the cross-Channel fleet, under the Belgian flag, for running between Dover and Ostend. This vessel, the *Princesse Elisabeth*, 350 feet long by 40 feet beam, was constructed at Antwerp, and successfully went through her trials on the Clyde, between the Cloch and the Cumbrae Lights, and maintained an average speed thereon of twenty-four knots on four runs. The vessel marks another step in advance made in turbine propulsion, and the speed obtained holds the record for this class of steamer. The hull of the vessel, boilers and parts of the machinery have been built by the John Cockerill Company; the turbine machinery was constructed at Wallsend, under contract with the Parsons Foreign Patents Company, Limited; the auxiliary machinery, wet and dry-air pumps, circulating, feed, oil, sanitary and bilge pumps, have all been supplied by Messrs. G. and J. Weir, Limited, of Glasgow.

Ferryboat Brooklyn—*Trial.*—The *Brooklyn* was the second to be tried of the four ferryboats built by the Maryland Steel Company for the City of New York.

The four ferryboats are the same size and built from the same plans. They are of the double-end type, having propeller at each end.

Length over all, 250 feet; length between perpendiculars, 246 feet; beam over guards, 66 feet; beam molded, 46 feet; depth, 19 feet 6 inches; draught, 12 feet; displacement at 11 feet 10 inches; draught, 1,394 tons.

Engines.—There are two compound engines, set with the L.P. cylinders adjacent, the crank shafts being coupled between the L.P. cylinders, and the shafting continues from end to end of the vessel.

The engines were designed for a working pressure of 175 pounds at the H.P. valve chest. Designed piston speed, 750 feet. The engines are same size, $\frac{22\frac{1}{2} \text{ ins.} \times 50 \text{ ins.}}{30.}$

The back frames of engines are cast and of inverted Y type. The front columns are forged. The engines are simple, easily accessible and substantial in their design and construction.

Boilers.—There are four Babcock & Wilcox boilers, located two in compartment forward and two in compartment abaft engine compartment.

Total G. S., square feet	340
H. S., square feet	12,500
Working pressure, pounds	225

The pressure is reduced to 175 pounds at the engine.

Condenser.—There is one cylindrical built-up condenser with shell of $\frac{3}{8}$-inch steel plate.

Diameter of tubes, inch	$\frac{3}{4}$
Thickness of tubes, B.W.G.	18
Total cooling surface, square feet	4,500

The cooling water makes a double pass, entering the top and returning through the bottom tubes.

This arrangement gives warmer feed water and requires less circulating water than where the cooling water passes through the bottom tubes first and leaves the condenser from top nest of tubes.

Pumps.—There is one centrifugal circulating pump supplying the condensing water. It is fitted to take water from a side, a bottom and a bilge injection. The bottom injection is to be used when the floating ice prevents the use of the side injection.

There is one twin air pump of Cameron make $\frac{10'' \times 22''}{18''}$.

There are two main feed pumps, one in each fireroom, and an auxiliary feed pump in the engine room. These pumps are of the Cameron vertical simplex type, the main feed pumps being $\frac{10'' \times 6''}{12''}$ and the auxiliary $\frac{12'' \times 6''}{16''}$. Besides the feed pumps there is a $2\frac{1}{2}$-inch Metropolitan injector in each fireroom for feeding the boilers.

The auxiliary feed pump alone was in use to supply water

to all boilers during the trial. It did this work easily and quietly, making 26 double strokes per minute.

There is one Cameron vertical simplex sanitary pump $\frac{6'' \times 8''}{8''}$, and one Cameron duplex fresh-water pump $\frac{4\frac{1}{2}'' \times 3\frac{3}{4}''}{4''}$.

There is a $\frac{12'' \times 9\frac{3}{4}''}{10''}$ Blake fire and bilge pump in the engine room.

Propellers.—There are two right-hand, four-bladed propellers.

Diameter, feet	11
Pitch, feet and inches	14- 4
Helicoidal area, square feet	47.66
Projected area, square feet	39.55
Ratio of projected area to helicoidal area	1 : 1.23
helicoidal area to disc area	.501

The trial was made in the Chesapeake Bay. The averages are as follows:

Steam pressure at engines, pounds	176
Recorded pressure (gauge), pounds	29
Vacuum, inches	24.3
Revolutions per minute, main engines	147.5
air pump	46.4
circulating pump	275
I.H.P., both engines	4,093
Speed in knots	15.93
Slip of propeller, per cent	24.3
Draught on trial, feet and inches	11-10
Displacement, tons	1,394
Wetted surface, square feet	9,066
I.H.P. per 100 square feet of wetted surface at speed of 10 knots	11.25
Admiralty coefficient	134.2

The boilers were operated under one-inch air pressure.

No account was taken of the weight of coal, so that the economical performance is not known. The operation, however, of the engines, boilers and all auxiliaries, as observed on the trial, was satisfactory.

OBITUARY.

LIEUTENANT COMMANDER ROBERT I. REID, U. S. N. (RETIRED).

It is with extreme regret that the Council announces the death of one of its members since the last issue of the JOURNAL.

Lieutenant Commander Robert I. Reid, U. S. N. (Retired), died at Caldwell, N. J., November 10, 1905.

BOOKS RECEIVED.

DAVID G. FARRAGUT.—By JOHN RANDOLPH SPEARS, published by GEORGE W. JACOBS & COMPANY, Philadelphia, Pa.

This is the third of the American Crisis Biographies, and gives in a comprehensive and interesting manner a description of those actions, both before and during our Civil War, in which Admiral Farragut took such a conspicuous part.

NOTES OF THE SOCIETY.

A regular meeting of the Society was held at the Navy Department, October 3d, at which nomimations were made for Officers of the Society for the coming year, as follows:

For President.

Commander A. B. CANAGA, U. S. N.
Commander W. F. WORTHINGTON, U. S. N.
Lieutenant Commander B. C. BRYAN, U. S. N.

For Secretary-Treasurer.

Lieutenant Commander T. C. FENTON, U. S. N.
Lieutenant C. K. MALLORY, U. S. N., Retired.

For Members of Council.
(Three to be selected.)

Rear Admiral G. W. BAIRD, U. S. N., Retired.
Engineer-in-Chief C. A. MCALLISTER, U. S. R. C. S.
Commander W. M. PARKS, U. S. N.
Commander R. S. GRIFFIN, U. S. N.
Lieutenant Commander B. C. BRYAN, U. S. N.
Commander W. W. WHITE, U. S. N., Retired.
Lieutenant A. M. COOK, U. S. N.
Civil Engineer A. C. CUNNINGHAM, U. S. N.
Lieutenant JOHN HALLIGAN, JR., U. S. N.
Lieutenant H. T. WINSTON, U. S. N.

Voting slips have been forwarded to all the members, to be returned at once. The votes will be counted at the Annual Meeting of the Society, to be held in the Bureau of Steam Engineering, Navy Department, at 4.30 P. M., Thursday, December 28, 1905.

At a meeting of the Council, held in the Bureau of Steam Engineering, October 27, 1905, Lieutenant C. K. Mallory, U. S. N., tendered his resignation as Secretary-Treasurer, and Lieutenant Commander Theo. C. Fenton, U. S. N., was elected to succeed him.